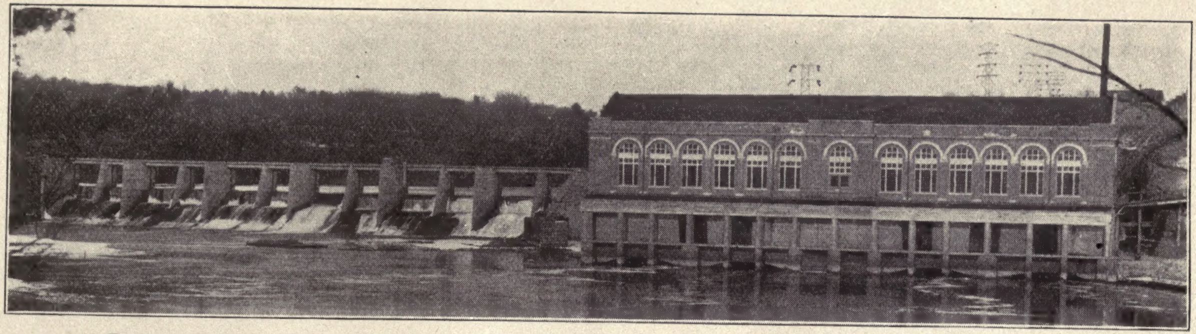
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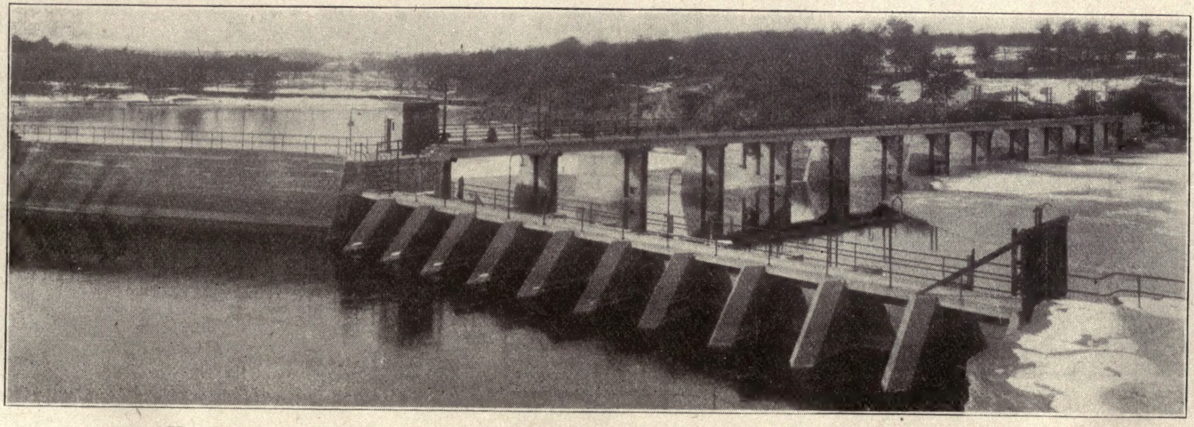
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WATER POWER ENGINEERING

THE THEORY, INVESTIGATION AND DEVELOPMENT OF WATER POWERS.

BY

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PREFACE TO THE SECOND EDITION.

In this edition the author has endeavored to bring the text into accord with the best modern (1915) practice.

Various changes, additions and rearrangements of the text have been made in order to treat the subject more fully and in a more logical order. The chapter on "Hydraulics" in the first edition was not found simple enough for the beginner or complete enough for the hydraulic engineer; it has therefore been omitted although some of the matters discussed therein have been added to other chapters. The chapters on "Rain-fall," "Run-off," and "Stream Flow," of the first edition, have also been omitted as not sufficiently complete for the purpose and with the intention of discussing same more fully in a volume on "Hydrology," now in course of preparation. In Chapter VI, the discussion of the hydrograph as a basis for power development has been somewhat extended.

The two systems of graphical turbine analysis, published in the first edition, are believed to be of considerable value. These methods were at the time so new that their advantages were not recognized, and they were overlooked in all reviews of the work. Since that time one or the other of these systems has been adopted by most turbine manufacturers for presenting their turbine data, and while they have also been utilized by some engineers, the advantages of these methods have not yet been fully appreciated by water power engineers. This has been partially due to the somewhat unsatisfactory treatment of the subject in the first edition. In this edition this subject has been more simply and completely treated in Chapters XI and XIII, and it is believed these methods will now be readily understood. The symbols for turbine coefficients have also been so changed as to be more suggestive of the functions they represent. Almost every chapter in the book has been revised and extended. Greater care has been given in the selection and extension of the literature following each chapter, and many foreign references not readily accessible and many references of doubtful value, have been eliminated, while references to much of the best literature which has appeared in the last seven years has been added.

In the seven years that have elapsed since the publication of the first edition of this work, there has been a very decided improvement in

both the capacity and the efficiency of low head turbines, and tests of turbines published in the appendix show the results attained in some of the best and latest turbine designs.

No attempt has been made in this work to treat in detail the subject of turbine design, as this work is intended, not for the turbine designer, but for the student and engineer who may be called upon to select turbines in connection with water power developments. While a knowledge of turbine design would be of advantage to the water power engineer, there are few engineers who have either the time or the capacity to become experts on all phases of this subject, and it has been the purpose of the author to cover these matters only to the extent that is considered essential for a fundamental knowledge of water power engineering. A much further investigation of all the subjects treated in this work is recognized as not only desirable but essential for the best results.

The author wishes to acknowledge the assistance of Mr. L. R. Balch who has given several months of his undivided attention to the revision of this work, also to Mr. H. L. Garner and other members of the author's office force. He also acknowledges many valuable suggestions from Professor S. M. Woodward, and from Mr. C. V. Seastone. At the suggestion of Mr. B. F. Groat, the writer adopted, in the first edition, the expression $\mathfrak{P} = \frac{n^2 P}{h^{\frac{5}{2}}}$ for specific power, instead of Bacheus' expression for specific speed $N = n \frac{\sqrt{P}}{h^{\frac{5}{4}}}$. This change has been found by the author of considerable advantage in the selection of turbines for water power work, and its origin should have been acknowledged in the first edition.

D. W. M.

Madison, Wis., Sept., 1915.

PREFACE TO THE FIRST EDITION.

In the development of a water power project the engineer is frequently called upon to do more than design and construct the power plant. He may be required to report on the adequacy of the supply, the head and power available and the probable variations in the same, the plan for development, the cost of construction and operation, and the advisability of the investment. A study of the entire project, therefore, becomes essential, and each factor must be carefully considered in detail to assure ultimate success. Each of the features of the development is of equal importance to the commercial success of the project. The majority of the failures in water power development have occurred from causes other than structural defects, and a knowledge of these other important and controlling factors is therefore quite as essential as a knowledge of design and construction. It must be said, however, that in respect to some of these controlling factors, practice has not been what it should be. This has resulted in many over-developments and illy advised installations, from which the power generated has not been equal to that anticipated, and in many poor financial investments amounting frequently to practical failures. The engineer has given much attention to design and construction but too little attention to the other fundamental considerations mentioned above on which the success of the project depends to an equal extent.

In the preparation of this book the author has endeavored to consider, briefly at least, all fundamental principles and to point out the basis on which successful water power development depends. The method of study and investigation outlined herein was developed by the author during twenty-five years of professional practice and in his efforts to illustrate the principles underlying the subject in his lectures to the senior class in water power engineering at the University of Wisconsin. A somewhat extended acquaintance with the literature relating to water power engineering leads the author to believe that in a number of features the principles and methods described in this book are somewhat in advance of present practice.

In practice, the hydraulic engineer, to determine the extent of a proposed hydraulic development, commonly depends on a study of the monthly averages of stream flow and of observed maximum and minimum flows. He usually assumes from his previous knowledge and

study that the development should be based on a certain minimum or average stream discharge per square mile of drainage area. The value of this method depends on the breadth of the engineer's local knowledge of rain-fall and run-off relations. With a sufficient knowledge of these conditions, this method may form a safe basis for water power development but it fails to give the complete information which is essential for a full comprehension of the subject. In other cases the development is predicated on a single, or on a very few, measurements of what is believed, or assumed to be, the low water flow of the stream. This method, even when accompanied by careful study of rain-fall records, is a dangerous one to employ as many over-developed water power projects demonstrate.

In practice the head available is usually determined for average conditions, or, perhaps, occasionally for low, average and high water conditions, and no detailed study of the daily effect on power is attempted. In Chapters VI and VIII this subject is presented in detail and a method of the investigation of this important subject, under all conditions of flow and all conditions of use, is outlined.

On the basis of the knowledge gained from the study of flow and head, the study of the power that can be developed for each day in the year and during each year for which actual or comparative hydrographs are available, is outlined. A study of the effect of pondage on power, a most important matter, though not always carefully considered, or appreciated, is also discussed in considerable detail in Chapters VI, VII and VIII.

In the selection of turbines for a water power project, the practice has been for the engineer, while drawing certain conclusions from the tables of manufacturers' catalogues, to present to the manufacturer the conditions under which the power is to be developed and to rely largely or entirely on the manufacturer for advice as to machinery to be used. In such cases he is dependent for results on guarantees which are usually quite indefinite in character and seldom verified by actual tests under working conditions, before the wheels are accepted and paid for. This has resulted in many cases in the installation of wheels which are entirely unsuited to the particular conditions under which they are installed.

Practical turbine analysis has not been treated except in the most general way in any publications except the various German treatises on the turbine in which the subject is discussed from the basis of turbine design. The author has developed the method of turbine an-

alysis and selection, outlined in Chapters XI and XIII, which applies to all wheels when tests of wheels of the series or type considered are available. These methods are based on the practical operating conditions of actual tests and are both theoretically and practically correct. The engineer should be able to intelligently select the turbines needed for the particular conditions of his installation and to determine, with a considerable degree of accuracy, the results on which he can depend during all conditions of head and flow.

It is believed that this treatment of the subject is sufficiently complete to place the selection of turbines on a better footing and that, when adopted, it will lead to the selection of better and more improved designs and assure more satisfactory results.

The subject of turbine governing has, for electrical reasons, become an important one. While a number of important papers have appeared on this subject, there is, so far as the author knows, no discussion in English which supplies the engineer a basis for a complete consideration of this subject. Chapter XIV, on the principles of turbine governing offers, it is believed, suggestions for the consideration of this subject which may prove of value to water power engineers.

The report on a water power project should involve a careful and complete investigation of the entire subject, and should be based on the broadest considerations of the project in all its relations. Many reports which have come to the author's attention have been too limited in scope and have included only general opinions which have not, to his mind, been sufficiently specific or based on sufficient information to warrant approval without extended investigations. In Chapter XXIV the author has outlined his idea of the extent and scope of such investigation and report, which he believes is essential for an intelligent investigation and a reliable opinion on this subject.

ACKNOWLEDGMENTS

There can be little which is strictly new or original in any technical work, and in offering this book to the profession, the author wishes to acknowledge his indebtedness to the large number of technical articles that have already appeared on various phases of the subject. Many references to such literature have been given at the end of the various chapters.

Many illustrations have been taken, with more or less change, from Engineering News, Engineering Record, Cassier's Magazine and Elec-

trical World and Engineer. Various manufacturers have furnished photographs and, in some cases, cuts of their wheels, governors and apparatus, in connection with which their names appear.

The author has been greatly aided by his assistants, both of his own private office and of the University staff. He wishes especially to acknowledge the assistance of Mr. L. F. Harza to whom Chapter XIV on The Speed Regulation of Turbine Water Wheels is largely due. Mr. Harza has also been of much assistance in the editorial work of publication. Especial acknowledgment is also due to Professor G. J. Davis, Jr., for the preparation of the diagram of Bazin's coefficients, etc. Mr. Robert Ewald assisted in the selection of material for illustrations, in the investigation of German literature, and the preparation of various graphical diagrams, including the first development of the characteristic curve.

The author also desires to acknowledge his indebtedness to Mr. C. V. Seastone, for advice and assistance in the arrangement of many of the chapters in this work and assistance in the editorial work of publication.

The sources of various other tables, illustrations, etc., are acknowledged in their proper places. D. W. M.

Madison, Oct. 1, 1908.

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WATER POWER ENGINEERING

CHAPTER I

INTRODUCTION

THE HISTORY OF WATER POWER DEVELOPMENT

1. **Early Development of Water Power.**—Most methods of power generation can be traced to an origin at no very remote period. Their development has been within historic times. The first development of water power, however, antedates history. Its origin is lost in remote antiquity.

Air and water, both physical agents most essential to life, have always been the most obvious sources of potential energy and have each been utilized for power purposes since the earliest times. Beside the Nile, the Euphrates, and the Yellow Rivers, thousands of years ago the primitive hydraulic engineer planned and constructed his simple forms of current wheels and utilized the energy of the river current to raise its waters and irrigate the otherwise arid wastes into fertility. Such primitive wheels were also utilized for the grinding of corn and other simple power purposes. From these simple forms and primitive applications have gradually been developed the modern water power installations of to-day.

2. **The Earliest Type of Water Wheel.**—The crude float wheel driven directly by the river current developed but a small portion of the energy of the passing stream. The Chinese Nora, built of bamboo with woven paddles, is still in use in the east (see Fig. 1, page 2), and was probably the early form of development of this type of wheel. The type is by no means obsolete for it is yet used for minor irrigation purposes in all countries. These wheels, while inefficient, served their purpose and were extensively developed and widely utilized. One of the greatest developments of which there is record was the float wheel installation used to operate the pumps at London Bridge for the first water supply system of the city of London, and constructed about 1581 (see Fig. 2, page 2). In all such wheels the paddles dip into the unconfined current which, when impeded by the wheel, heads up and passes around the sides of the wheel and thus allows only a small part of the current energy to be utilized.

3. The Undershot Wheel.—The introduction of a channel confining the water and conducting it to a point where it could be applied directly to the undershot wheel, was an improvement that permitted

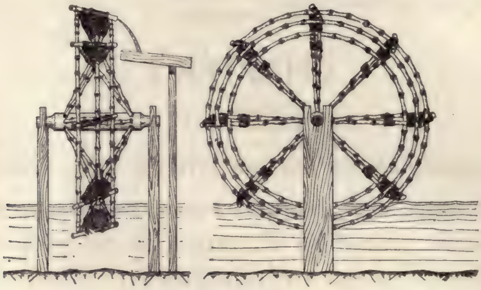


Fig. 1.—Chinese Nora or Float Wheel Used From Earliest Times to Present (see page 1).

the utilization of about thirty per cent. of the theoretical power of the water. This form of water wheel was most widely used for power development until the latter half of the eighteenth century.

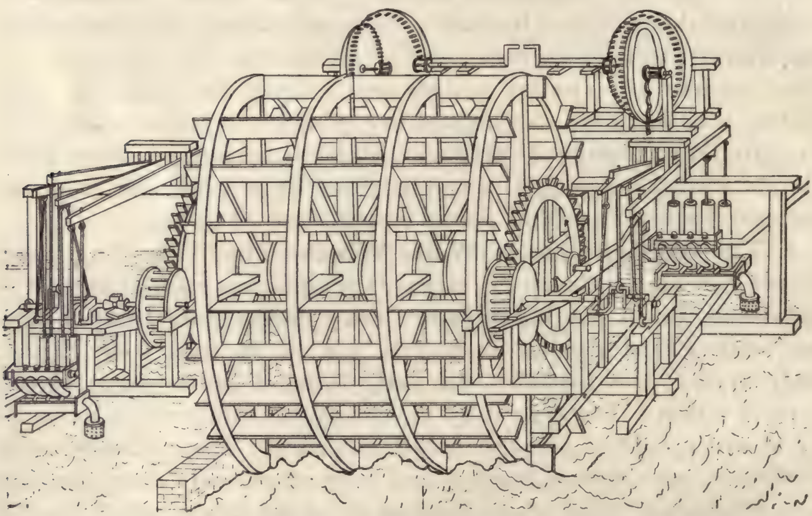


Fig. 2.—Float Wheel Operating Pumps for Water Supply of London 1581 (from Matthews' Hydraulia Lond. 1835). (See page 1.)

In the float and undershot wheels the energy of water is exerted through the impact due to its velocity. The heading up of the water, caused by the interference of the wheel, results also in the exertion of pressure due to the weight of the water, but this action

has only a minor effect. The conditions of the application of the energy of water through its momentum is not favorable to high efficiency in this type of wheels and the determination of this fact by Smeaton's experiments undoubtedly was an important factor in the introduction and adoption of the overshot water wheel.

4. The Overshot and Breast Water Wheel.—In the overshot water wheel the energy of water is applied directly through its weight by the action of gravity, to which application the design of the wheel is readily adapted. Such wheels when well constructed have given efficiencies practically equal to the best modern turbine,

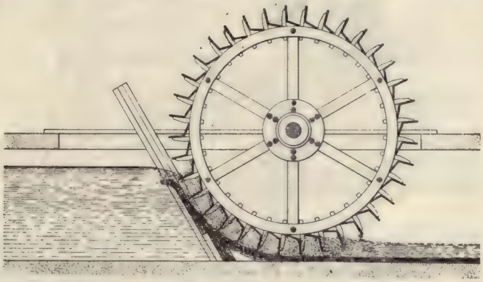


Fig. 3.—Breast Wheel Used From About 1780 to About 1870.

but on account of their large size and the serious effects of backwater and ice conditions, they are unsatisfactory for modern power plants (see Fig. 11, page 10).

Following the work of Smeaton, the breast wheel (see Fig. 3) was developed in England largely through the work of Fairbairn and Rennie. The latter in 1784 erected a large wheel of this type to which he applied the sliding gate from which the water flowed upon the wheel instead of issuing through a sluice as formerly. About this time the fly-ball governor, which had been designed and adapted as a governor for steam engines by Watt, was applied to the governing of these wheels and by means of these governors the speed of the wheel under varying loads was kept sufficiently constant for the purpose to which they were then applied (see Fig. 4, page 4).

Another mode of applying water to wheels under low falls was introduced by M. Poncelet (see Fig. 5, page 4). Various changes and improvements in the form of buckets, in their ventilation so as to permit of complete filling and prompt emptying, and in their structure, took place from time to time, and until far into the middle of the nineteenth century these forms of wheels were widely used for water power purposes.

5. The Development of the Turbine.—The invention of any important machine or device is rarely the work of a single mind. In general such inventions are the result of years of experience of many

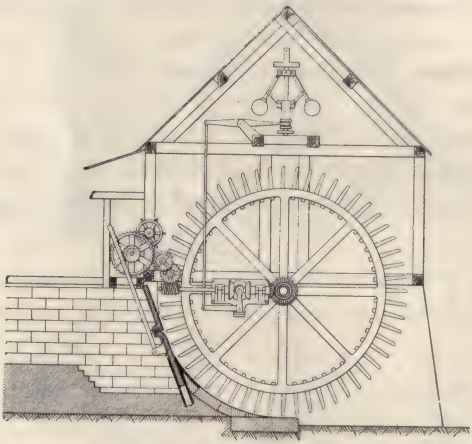


Fig. 4.—Breast Wheel About 1790 Showing Early Application of Governor (see page 3). (After Glynn.)

men which may be simply correlated by some designer, to whom often undue credit is given. To the man who has gathered together past experiences and embodied them in a new and useful invention

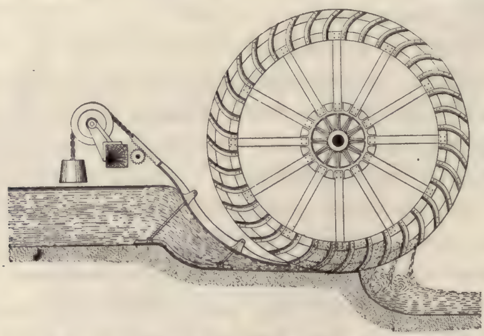


Fig. 5.—Poncelet's Wheel (see page 3).

and perhaps through whose energy practical applications are made of such inventions, the credit is frequently assigned for ideas which have been lying dormant, perhaps through centuries of time. Every inventor or promoter of valuable improvements in old methods and old construction is entitled to due credit, but the fact should nevertheless be recalled that even in the greatest inventions very few

radical changes are embodied, but old ideas are utilized and rearranged and a new and frequently much more satisfactory combination results. Improvements in old ideas are the improvements which are the most substantial. Inventions which are radically new and strictly original are apt to be faulty and of little practical value.

6. Fundamental Ideas of the Turbine.—The embryo turbine may be distinguished in the ancient Indian water mill (see Fig. 6). A

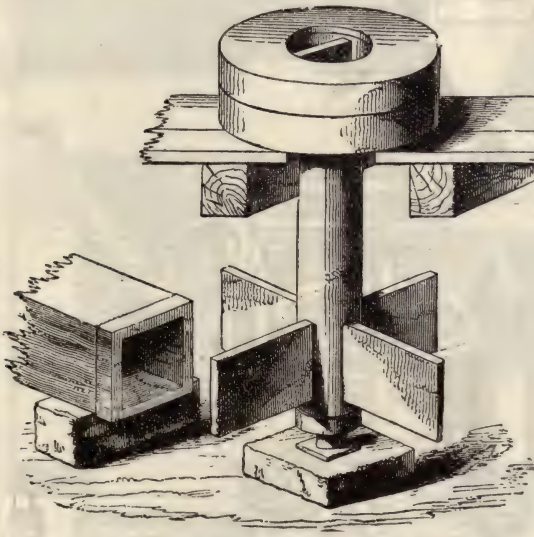


Fig. 6.—Ancient Indian Water Wheel (after Glynn). Containing Fundamental Suggestion of Both Turbine and Impulse Wheels.

similar early type of vertical wheel used in Europe in the sixteenth century, the illustration of which was taken from an ancient print (see Sci. Am. Sup. Feb. 17, '06) is shown in Fig. 7, page 6. Barker's mill in its original form or in the form improved by M. Mathon de Cour, embodied the principal idea of the pressure turbine, and was used to a considerable extent for mill purposes. In 1845 James Whitelaw suggested an improved form which was used in both England and Germany early in the nineteenth century (see Fig. 8, page 7). Many elements of the modern turbine were conceived by Benjamin Tyler, who received letters patent for what he termed the "Wry Fly" wheel in 1804. The description of this wheel as contained in the patent specifications is as follows:

"The Wry Fly is a wheel which, built upon the lower end of a perpendicular shaft in a circular form, resembles that of a tub. It

is made fast by the insertion of two or more short cones, which, passing through the shaft, extend to the outer side of the wheel. The outside of the wheel is made of plank, jointed and fitted to each other, doweled at top and bottom, and hooped by three bands of

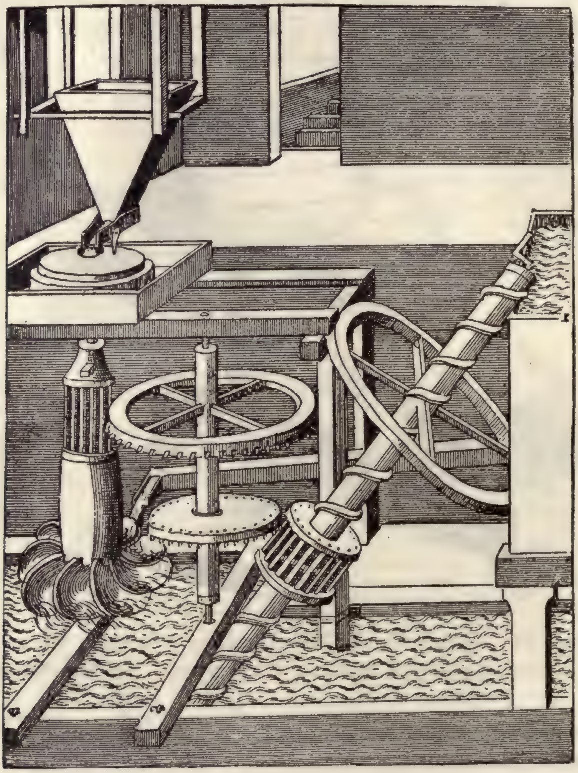
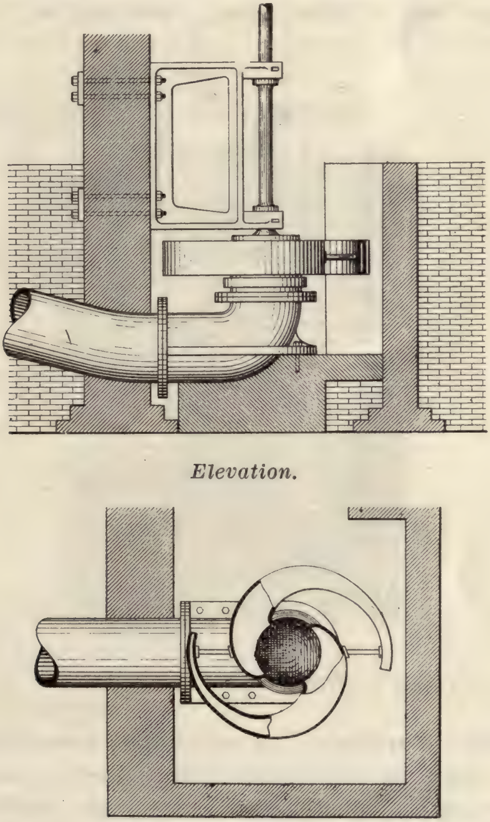


Fig. 7.—Early Vertical Wheel. Containing Fundamental Suggestion of the Turbine (see page 5).

iron, so as to make it water-tight; the top must be about one-fifth part larger than the bottom in order to drive the hoops, but this proportion may be varied, or even reversed, according to the situation of place, proportion of the wheel, and quantity of water. The buckets are made of winding timber, and placed inside of the wheel, made fast by strong wooden pins drove in an oblique direction; they are fitted to the inside of the tub or wheel, in such a manner as to form an acute angle from the wheel, the inner edge of the bucket inclining towards the water, which is poured upon the top, or upper

end of it about twelve and a half degrees; instead of their standing perpendicular with the shaft of the wheel they are placed in the form of a screw, the lower ends inclining towards the water, and against the course of the stream, after the rate of forty-five degrees; this, however, may be likewise varied, according to the circumstances of the place, quantity of water, and size of the wheel."



Elevation.

Plan and Partial Section.

Fig. 8.—Early Vertical Wheel. Containing Fundamental Suggestion of the Turbine (see page 6). (After Glynn.)

From the description it will be noted that, with the exception of the chutes, the principal features of the modern turbine were here anticipated. The "Wry Fly" wheel was an improvement on the "tub" wheel which was then in use to a considerable extent in the country.

These various early efforts received their first practical consummation and modern solution through various French inventors early in the nineteenth century. The "Roue à Cuves" (Fig. 9) and the "Roue Volant" (Fig. 10, see page 9) had long been used in France, and were the subject of extensive tests by MM. Piobert and Tardy at Toulouse. Those various wheels received the water tangentially through an opening or spout, being practically an improvement on the old Indian mill by the addition of a rim and the modification of the form of buckets.

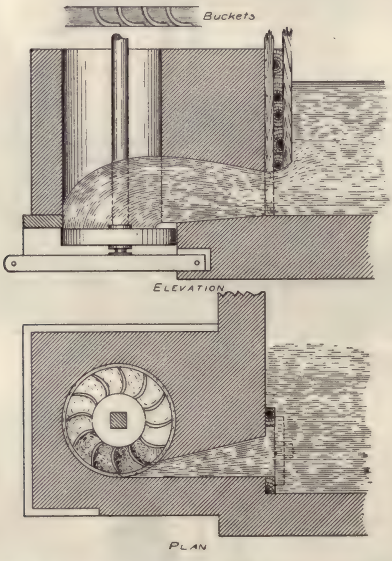


Fig. 9.—Roue à Cuves (after Glynn).

7. **The Modern Turbine.**—The next improvement consisted in the addition of a spiral or scroll case to the wheel, by means of which the water was applied equally to all parts of the circumference passing inward and downward through the wheel. To the French inventors, Koechlin, Fourneyron and Jonval, is largely due the design of the turbine in a more modern and practical form. By the middle of the nineteenth century these wheels had met with wide application in France and been adopted and considerably improved by American and German engineers, but were scarcely known in England (see "Power of Water," by Jos. Glynn, 1852). The turbine was introduced into the United States about 1843 by Ellwood Morris, of Pennsylvania, but was developed and brought to public attention

more largely through the inventions of Uriah A. Boyden, who in 1844 designed a seventy-five horse-power turbine for use at Lowell, Mass. (see Fig. 118, page 217). The great advantage of the turbine over the old style water wheel may be summarized as follows: (see Figs. 11 and 12, page 10).

First: Turbines occupy a much smaller space.

Second: On account of their comparatively high speed they can

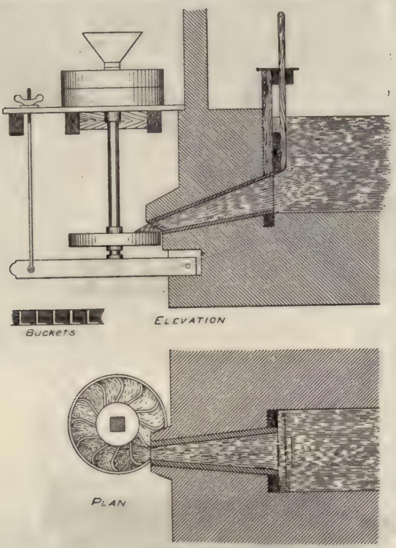


Fig. 10.—Roue Volant (see page 8). (After Glynn.)

frequently be used for power purposes without gearing and with a consequent saving in power.

Third: They will work submerged.

Fourth: In consequence of the ability to work "submerged" the turbine can efficiently utilize considerable variations in head, to which condition the old style water wheel is not applicable.

Fifth: Turbines may be utilized under almost any head or fall of water. They have been used under heads as low as sixteen inches and as high as 670 feet.

Sixth: Turbine water wheels are built of much greater capacity than is practicable with overshot wheels.

Seventh: By means of turbines, water powers of much greater magnitude can now be developed than would be possible with the older classes of water wheels.

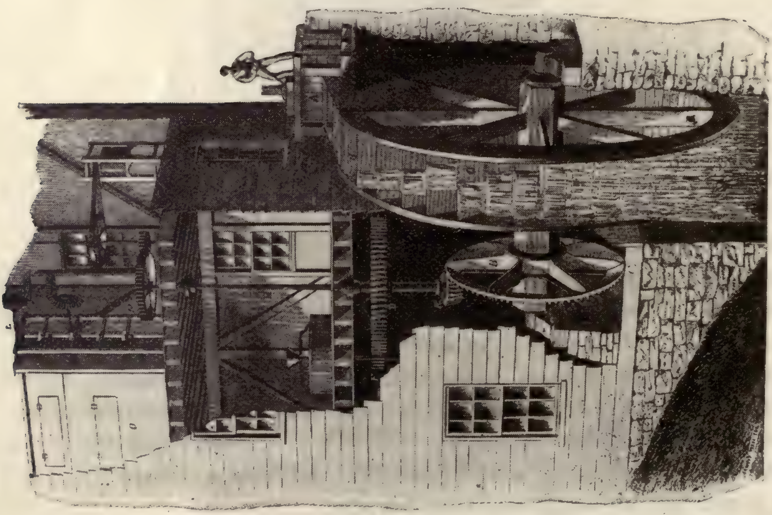


Fig. 11.—Installation of Overshot Wheel.

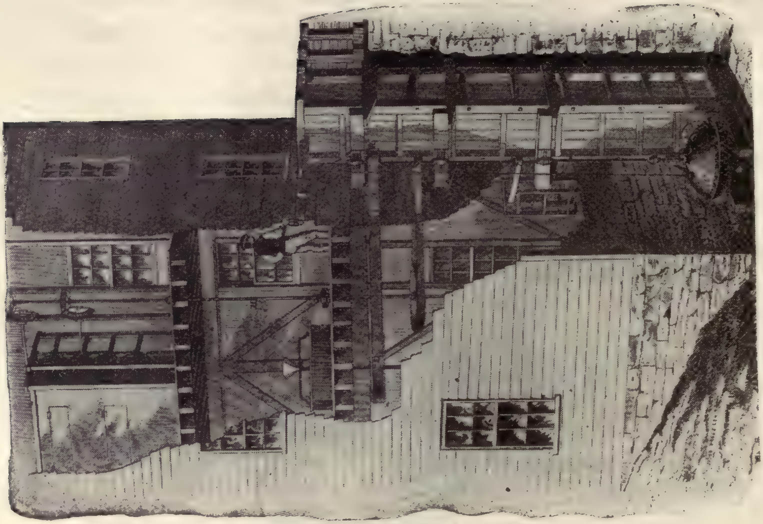


Fig. 12.—Installation of Turbine.

Comparative Installation of Water Wheel and Turbine (see page 9). (After Grimshaw.)

Eighth: Turbines are more readily protected from interference of ice.

8. The American or Francis Turbine.—Through the efforts of Uriah A. Boyden and James B. Francis (1849), the Fourneyron turbine became the leading wheel in New England for many years.

In 1838 Samuel B. Howd of Geneva, New York, patented the "inward flow" wheel, in which the direction of flow in the Fourneyron turbine was reversed. This seems to have been the origin of the

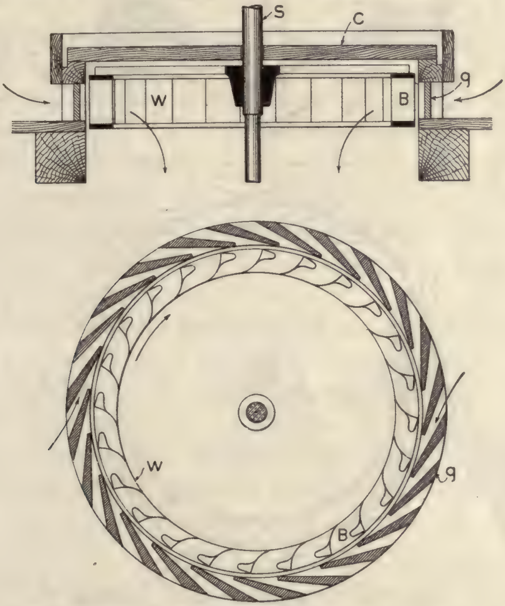


Fig. 13.—Inward Flow Wheel by S. B. Howd (see page 12). (After Francis.)

American type of turbine, and the Howd wheel was followed by a large number of variations of the same general design on which American practice has been based for many years. About 1849, James B. Francis designed an inward flow turbine of the same general type as the Howd wheel. Two of these wheels were constructed by the Lowell Machine Shop for the Boott Cotton Mills. In the "Lowell Hydraulic Experiments" page 61, Mr. Francis refers to the previous patent of Howd and says: "Under this patent a large number of wheels have been constructed and a great many of them are now running in different parts of the country. They are known in some places as the Howd wheel, in others as the United States wheel. They have uniformly been constructed in a very simple and

cheap manner in order to meet the demands of the numerous classes of millers and manufacturers who must have cheap wheels if they have any."

Figure 13, page 11, shows a plan and vertical section of the Howd wheels as constructed by the owners of the patent rights for a portion of the New England states. In this cut *g* indicates the wooden

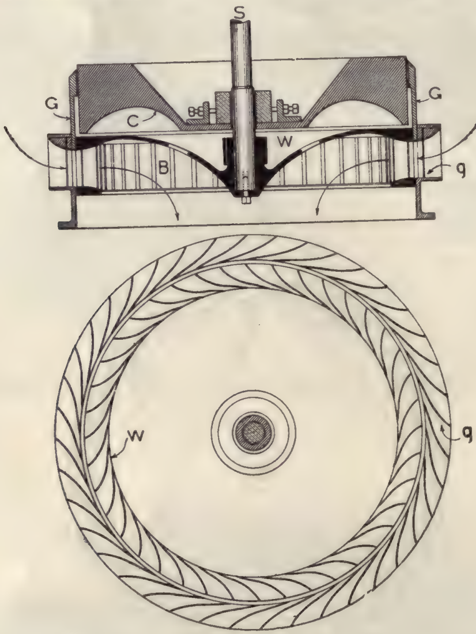


Fig. 14.—Original Francis Turbine.

guides by which the water is directed on to the buckets; *W* indicates the wheel which is composed of buckets of cast iron fastened to the upper and lower crowns of the wheel by bolts. The upright crown is connected with the vertical shaft *S* by arms. The regulating gate is placed outside of the guides and is made of wood. The upright shaft *S* runs on a step at the bottom (not shown in the cut). The projections on one side of the buckets, it was claimed, increased the efficiency of the wheel by diminishing the waste of the water.

The wheel designed by Francis was on more scientific lines, of better mechanical construction (see Fig. 14) and is regarded by many as the origin of the American turbine. The credit of this design is freely awarded to Francis by German engineers, this type of wheel being known in Germany as the Francis Turbine.

This wheel as originally constructed by Francis had less comparative power and speed than the Boyden-Fourneyron turbine (see Tables 1 and 2, page 14).

The Francis wheel was followed by other inward flow wheels of a more or less similar type. The Swain wheel was designed by A. M. Swain in 1855. The American turbine of Stout, Mills and Temple (1859), the Leffel wheel, designed by James Leffel in 1860, and the Hercules wheel, designed by John B. McCormick in 1876, are among the best known and earliest of the wheels of this class.

By 1870 the turbine had largely superseded the water wheel for manufacturing purposes at the principal water power plants in this country. The old time water wheel has since become of comparatively small importance, but it is still used in many isolated places where it is usually constructed by local talent, and adapted to local conditions and necessities.

One or two companies have, within recent years, begun the manufacture of steel overshot wheels which have been used successfully for the development of small powers.

The current wheel is still widely used for irrigation purposes and in many instances is a useful and valuable machine.

9. Modern Changes in Turbine Capacity.—A radical change has taken place in later years in the design of turbines for low head power developments. The adoption of deeper, wider and fewer buckets has resulted in a great increase in the comparative power and speed of wheels of later design. This development may be seen by an examination of Table 1 (see page 14) in which are compared the power and speed of various wheels of the same diameter, designed at various dates from 1849 to 1914. This table shows that the more recent designs have a capacity nearly twenty-two times as great as the original Francis turbine of 1849.

Table 2 (see page 14) shows the development in the hydraulic turbine and gives the comparative diameter and speed of various wheels designed at various dates from 1849 to 1914, and capable of delivering the same horse power under twenty-five foot head. This table shows that the more recent designs have a speed nearly six times as great as the original Francis turbine of 1849.

To appreciate these tables, it is necessary to understand that large capacity and high speed are desirable qualities of the turbine only for certain definite purposes and under certain definite conditions. Under other conditions, wheels of lower capacity and lower speed

TABLE 1.

Comparative Power and Speed of Various Turbines of 30" Diameter Under 25' Head Showing Increase in Power of Wheels of Recent Design. Results Calculated From Commercial Tests of Wheels.

Turbine	Reference	When Tested	Horse Power	Relative Power	Power Coefficient "p"	R.P.M.
Boott (Francis Design)...	W. P. E. Page 703	1849	25.2	1.0	.000224	198
Fourneyron (Francis De'n)	W. P. E. Page 707	1851	59.5	2.36	.000528	199
Leffel-Standard	Emerson Page 25	1869	63.2	2.51	.000562	242
American	Emerson Page 203	1873	63.3	2.51	.000563	203
Swain	Emerson Page 196	1874	85.5	3.38	.000760	233
Hercules	Emerson Page 229	1876	161.0	6.38	.001430	200
New American...........	Emerson Page 552	1894	178.0	7.07	.001580	232
Leffel-Samson	Holyoke Test 979	1897	213.0	8.43	.001890	260
Modern Turbine..........	Holyoke Test 1509	1904	220.5	8.75	.001960	242
Modern Turbine..........	Holyoke Test 1819	1909	253.0	10.00	.002250	228
Modern Turbine..........	Holyoke Test 1796	1910	317.0	12.55	.002820	247
Modern Turbine..........	Holyoke Test 2001	1911	360.0	14.30	.003200	248
Modern Turbine..........	Holyoke Test 2160	1912	394.0	15.60	.003500	240
Modern Turbine..........	Holyoke Test 2122	1912	427.5	16.95	.003800	250
Modern Turbine..........	Holyoke Test 2208	1913	478.0	19.00	.004250	233
Modern Turbine..........	Holyoke Test 2363	1914	552.5	21.90	.004910	250

TABLE 2.

Comparative Size and Speed of Various Turbines of 25 H. P. Under 25' Head Showing Decrease in Size and Increase in Speed of Wheels of Recent Design.

	Relative D^2	Relative R.P.M.	Diameter in Inches	R.P.M.
Boott (Francis Design).......	100.0	1.0	30.0	198
Fourneyron (Francis Design)..	42.4	1.54	19.55	305
Leffel-Standard	39.6	1.94	18.9	384
American	39.6	1.62	18.9	322
Swain	29.5	2.16	16.3	428
Hercules	15.7	2.54	11.9	504
New American...............	14.2	3.11	11.3	616
Leffel-Samson	11.8	3.82	10.3	758
Modern Turbine..............	11.4	3.61	10.15	715
Modern Turbine..............	10.0	3.65	9.47	723
Modern Turbine..............	7.9	4.42	8.45	876
Modern Turbine..............	7.0	4.73	7.95	937
Modern Turbine..............	6.4	4.80	7.60	950
Modern Turbine..............	5.9	5.20	7.30	1030
Modern Turbine..............	5.3	5.12	6.90	1015
Modern Turbine..............	4.5	5.93	6.38	1174

are necessary to properly meet the conditions of operation and are therefore the best when such conditions prevail. Most turbine manufactures, therefore, now build various types of turbines in which the capacities and speeds vary well toward the limits shown in these tables and therefore cover, as completely as practicable, the range of conditions created by the demands of practical service.

10. **Early Development of Impulse Wheels.**—As previously noted (Figs. 6, page 5, and 7, page 6), water wheels of the impulse type were among the earlier forms used. In the practical construction of water wheels for commercial purposes in this country, the reaction turbine was, however, the earliest form of development. This was because the reaction turbine was best suited for the low heads first developed. As settlement advanced from the more level country into the mountainous regions the conditions were found to radically differ. In the former location large quantities of water under low heads were available; in the latter, the streams diminished in quantity but the heads were enormously increased. These conditions demanded an entirely different type of wheels for power purposes and the demand was met by the construction of the tangential wheel now so widely and successfully used in the high head plants of the West.

The earliest scientific consideration of impluse wheels in this country was by Jearum Atkins who, apparently, anticipated the design of the wheels of the Girard type in Europe by his design of such a wheel in 1853* (see Fig. 15, page 16).

In Atkins' first application for a patent (in 1853) he shows a clear conception of the principles of the impulse wheel.

After describing the mechanical construction of his wheel, Mr. Atkins says: "The important points to be observed in the construction of this wheel and appendages, are: First, that the gearing * * * should be so arranged as to allow the wheel's velocity at the axis of the buckets to be equal to one-half the velocity of the water at the point of impact, * * *

"As the power of water, * * * is measured by its velocity, * * * it is obvious that in order that the moving water may communicate its whole power to another moving body, the velocity of the former must be swallowed up in the latter. This object is effected by the before-described mode of applying water to a wheel in

* See "Tangential Water Wheels" by John Richards, Cassier's Magazine, vol. V, p. 117.

the following manner, the velocity of the wheel, as before stated, being one-half that of the water.

"Let us suppose the velocity of the water to be twenty-four feet per second; then the velocity of the wheel being twelve feet per second, the relative velocity of the water with respect to the wheel, or the velocity with which it overtakes the wheel, will be twelve feet per second. Now it is proved theoretically, and also demonstrated by experiment, that water will flow over the entire surface

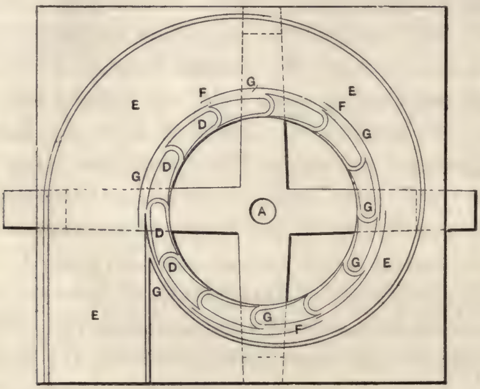


Fig. 15.—Plan of Atkins Wheel and Wheel Case (1853). (From Cassier's Magazine, Vol. V, p. 119.) (See page 15.)

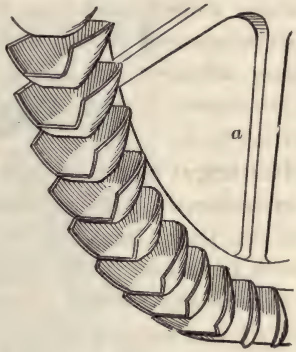
of the semi-circular buckets of the wheel with the same velocity with which it first impinged against them, or twelve feet per second. Then, as the water in passing over the face of the buckets has described a semi-circle, and as its return motion on leaving the wheel is in an opposite direction from that of the wheel, its velocity with respect to the wheel being twelve feet per second, and as the wheel has an absolute velocity of twelve feet per second, it is obvious that the absolute velocity of the water with respect to a fixed point is entirely suspended at the moment of leaving the inner point of the buckets, its whole velocity, and consequently its whole power, having been transmitted to the wheel."

Mr. Atkins' first application for a patent was rejected. After a long illness, from which he later recovered, he again applied for a patent which was finally granted in 1875. The Atkins patents are simply of historical interest as his inventions have had little effect on the practical development of the impulse wheel.

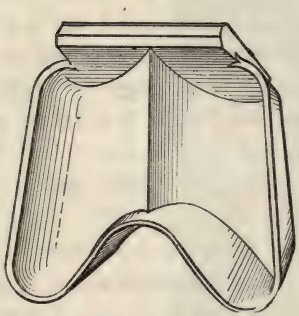
The impulse wheel found its earliest practical development in Cal-

ifornia where the conditions for the development of power made such a wheel necessary. The early tangential wheel, used on the Pacific Coast, was quite simple in construction and the development of the buckets, which began with the simpler flat and curved forms, was very largely based on the experimental method used for the development of the reaction turbine in the East. Experiments were

a. Moore Bucket, 1874.

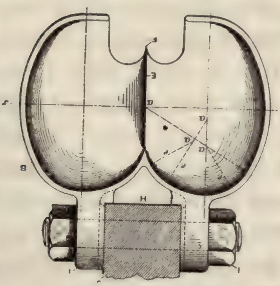


b. Knight Buckets, 1870.

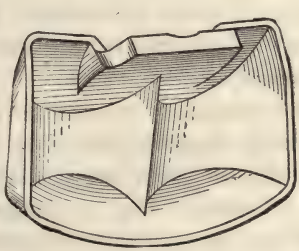


c. Dodd Bucket, 1889.

d. Hug Bucket, 1897.



e. Doble Ellipsoidal Bucket, 1889.



f. Pelton Bucket, 1880.

Fig. 16.—Buckets of Tangential or Impulse Water Wheels (Trans. Am. Inst. Mining Eng. 1899). (See page 18.)

made at the University of California, by Mr. Ralph T. Brown, as early as 1883, and the bulletin published by the department was the earliest literature on tangential wheels published in this country.

With the early development of the tangential bucket are connected the names of Knight, Moore, Hesse, Pelton, Hug, Dodd and Doble, and many other inventors, whose wheels have become well-known and widely used (see Fig. 16, page 17). The most extensive early development of this wheel was by The Pelton Water Wheel Company whose work has been so widely known and used as to make the name "Pelton Wheel" a common title for all wheels of the tangential type.

11. Historical Notes on Water Power Development.—Water mills were introduced at Rome about seventy years B. C. and were first erected on the Tiber. Vitruvius describes their construction as similar in principle to the Egyptian Tympanum. To their circumference were fixed floats or paddles which when acted upon by the current of the stream drove the wheel around. Attached to this axis was another vertical wheel provided with cogs or teeth. A large horizontal wheel toothed to correspond with it worked on an axis, the upper head of which was attached to the mill stone. The use of such water wheels became very common in Italy and in other countries subject to Roman rule.

Some of the early applications of water power are of interest. In 1581 a pump operated by a float wheel was established at London Bridge to supply the city of London with water (see Fig. 2, page 2). In 1675 an elaborate pumping plant driven by water wheels was established on the Seine river near Saint Germain. For this plant a dam was constructed across the river and chutes were arranged to conduct the water to the undershot water wheels. These were twelve or more in number, each operating a pump that raised the waters of the Seine into certain reservoirs and aqueducts for distribution.

The pumping of water for agricultural irrigation and drainage, domestic supplies and mine drainage, was undoubtedly the first application of water power, and still constitutes an important application of water. Fig. 17, page 19, from an article by W. F. Dupfee, published in Cassier's Magazine of March, 1899, illustrates a primitive application of the water wheel to the pumping of water from mines. Fig. 18, page 20, also shows the great Laxy overshot water wheel in the Isle of Man which is still used for mine drainage. The

wheel is about seventy feet in diameter and the water is brought from the hills a considerable distance for power purposes.

12. Development of Water Power in the United States.—In this country one of the first applications of water power was the old tidal mill on Mill Creek near Boston, constructed in 1631, which was followed by the extensive developments of small powers wherever

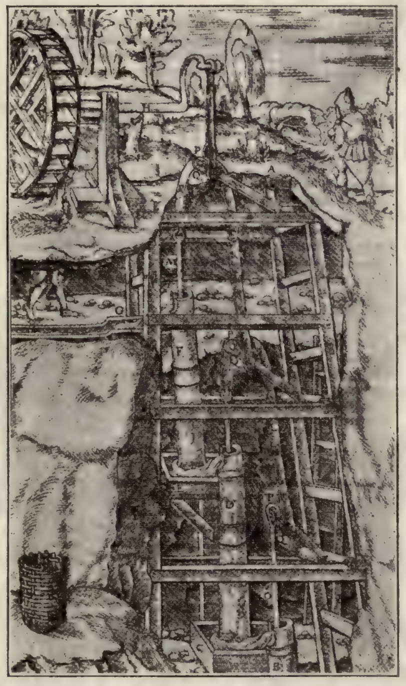


Fig. 17.—Early Application of Undershot Water Wheel to Mine Drainage, Date Unknown (see page 18). (From Cassier's Mag. March, 1899.)

settlements were made and water power was available. Often availability of water power determined the location of the early settlement.

About 1725 the first power plant was established along the Niagara River. This was a water-driven saw-mill constructed by the French to furnish lumber for Fort Niagara.

The last fifteen years have witnessed a somewhat rapid development of water powers. The increase in industries and the various demands for power and energy, the increased cost of coal, the improvement in electrical methods of generation and, more especially,

the rapid advance in the art of high tension long distance transmission have all united to accelerate this development. Water powers once valueless on account of their distance from centers of manufacturing and population are now accessible, and such powers are being developed and their energy brought into the market.

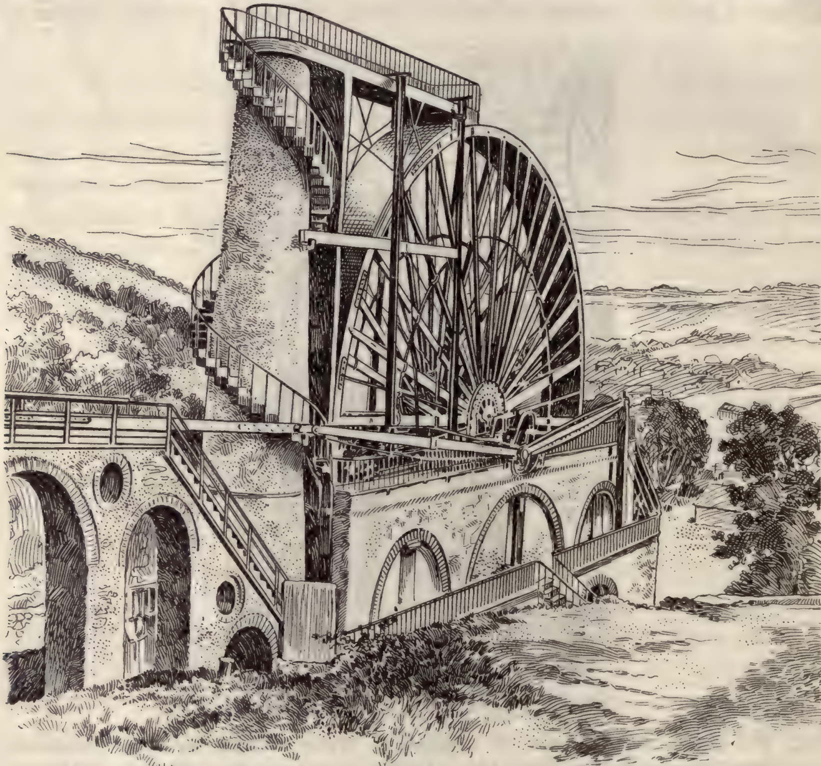


Fig. 18.—Laxy Overshot Water Wheel, Isle of Man (see page 18).

The statistics of the Bureau of Corporations (Report of the Commissioner of Corporations on Water Power Development in the United States, March 14, 1912) show the total developed water power was, in the year 1902, about 2,328,000 H. P. In 1907 this had increased to approximately 3,503,000 H. P., an increase of fifty per cent., and the developed power in June, 1911, was estimated at approximately 6,000,000 H. P., or an increase of seventy-one per cent. over that for 1907. The total developed water power has probably increased to date (July, 1915) to about 7,500,000 or 8,000,000 H. P.

Water power for central station and electric railway service increased from 487,000 H. P. in 1902 to 1,441,000 H. P. in 1907, or nearly 200 per cent. The increase in water power development among manufacturing industries for the years 1900 to 1905 was slightly more than eleven per cent., thus showing the large part that water power development plays at present in public service enterprises.

The Bureau of Corporations estimates the total potential water power in the United States (on the basis of seventy-five per cent. generation efficiency), at from 26,736,000 H. P. for the average minimum six months' flow, to 51,400,000 H. P. for the average maximum six months' flow of the rivers of the United States. Of the minimum power, approximately forty-three per cent. is found in the three states of California, Oregon, and Washington alone, and over seventy per cent. in nine of the western states.

From the foregoing it will be noted that the developed water power in June, 1911, was less than one-fourth of the estimated minimum potential water power of the United States, and it is further given that the developed water power was about twenty per cent. of the total installed stationary power in the United States.

13. **Conservation and Its Effect on Water Power Development.**—In recent years there has been a widely spread popular delusion in the public mind that, as undeveloped water power is energy going to waste, therefore, those who are developing and utilizing such power are garnering great wealth from a natural resource which justly belongs to the people of the states or of the nation, and from which they should receive benefit. If the energy of water could be turned into power without expense or hazard, there might be a legitimate reason for such an opinion. Such a result, however, can never be obtained in the development of any natural resource. A large proportion of the value derived from any resource is obtained from the invested capital and the resulting fixed and operating expenses; the undeveloped resource itself has a comparatively small value. Investments in water power and in the development of other natural resources are frequently expensive failures, and, while in a few cases the returns from such investments may be unduly large, the percentage of such cases is small. There is no line of development of natural resources so universally safe that such development must not be regarded as largely speculative and subject to many risks and contingencies.

The liberal water power laws which were in force prior to conservation agitation gave rise to the rapid development of water powers, which were not, however, always financially successful. During the last few years such development has been greatly restricted by subsequent unfavorable legislation. It has been held by some that the United States and the states have made a mistake in disposing of the ownership in fee of water power lands, and the idea has been advanced that in the future the nation and the state should reserve the ownership of such property and simply lease, under proper restrictions, the development of these natural resources by private enterprise. This principle is one of policy only, and would appear to be wise and generally beneficial if carried out under just restrictions, and would harm no one where the ownership of these resources really lies in the government. In some cases this idea has been extended, and a legislative attempt has been made to virtually confiscate water power property, the fees in which have already passed into private ownership, or to appropriate its value for the public benefit.

The most valuable and accessible water powers were largely developed before adverse legislation began and it is undoubtedly true that if legal restrictions to the development of the remaining water power of the states and of the nation were entirely removed, their value in many cases is so limited by natural and commercial conditions that the development of power would be slow. The great majority of the potential water powers of the nation have, undoubtedly, no present value whatever, and only a remotely speculative value for possible developments in the future. Most thoughtful men will agree on the advisability of reasonable restrictions on the development of water powers that are actually owned by the nation or the states; but few will agree to the policy of the establishment of legal restrictions to the development of potential power owned by the states or the nation, which will not take into account the hazard of the investment and the reasonable protection of the property of the parties who may undertake such development, or to the policy apparently adopted in some states of needlessly hampering the owners of water powers which have already passed to private ownership. Water powers can be conserved only by actual use, and while the corporations developing the same may and should realize a profit, it is well to remember that an enormous profit will also be realized by the public through the judicious expenditure of the vast

sums necessary for their development. The consequent increase in land values, and the values of other properties, the increase in population and in the value of all taxable property, the conservation of millions of tons of coal per annum for future generations, and the substantial development of the country adjacent to the developed water powers are public benefits commonly unconsidered, but greatly superior to any public interest which the states or the nation can possibly have in the undeveloped water power resources.

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CHAPTER II

POWER

14. Symbols Used in This Chapter.—The letters and symbols used in this chapter have the following significance:

a = Area (in square inches) against which pressure is exerted.
E = Total energy available.
E′ E″ E‴ = Energy losses in friction, leakage, velocity or other ways, in any machine or system of machines.
g = Acceleration due to gravity (32.2 feet per second per second).
h = The total available head in feet.
P = Horse power = 550 lbs. raised one foot per second against gravity = 550 foot pounds per second.
q = Quantity of water in cubic feet per second.
s = The space (in lineal feet) through which the area under pressure moves.
t = Time in seconds.
W = Total weight of water.
w = Weight of a cubic foot of water (practically 62.5 lbs.).
v = The velocity of flow (in lineal feet per second).

15. The Development of Potential Energy.—The development of natural sources of potential energy, the transformation of such energy into forms which can be utilized for power, and its transmission to points where it can be utilized for commercial purposes, constitutes a large portion of the work of the engineer. The water power engineer primarily deals with energy in the form of flowing or falling water, but his knowledge must extend much further for he encounters various other forms of energy. Some of the energy available from the potential source will be lost by friction in bringing the water to and taking it from the wheel. Some is lost in hydraulic and mechanical friction in the wheel; additional losses are sustained in every transformation, and, if electric or other forms of transmission are used or auxiliary power is necessary for maintaining continuous operation, the engineer will be brought in contact with energy in many other forms.

It should be the engineer's purpose to so select or design the mechanism or machines that he installs, that as great a proportion of energy from the available source shall be utilized as may be found practicable in any given case.

16. Expression for Energy.—Mechanically, energy is the exertion of force through space. The amount of available energy of water that may be theoretically utilized is measured by its weight (the force available) multiplied by the available head (the space through which the force may be exerted).

$$E = Wh \tag{1}$$

From equation (1) it will be noted that the energy of water is in direct proportion to both the head and quantity. This energy may be exerted in three ways which may be regarded as more or less distinct but which are usually exercised, to some extent at least, in common. The exertion of this energy in these three ways expressed in terms of horse power, are as follows:

First: By its weight which is exerted when a definite quantity of water passes from a higher to a lower position essentially without velocity. This method of utilization is represented by the equation

$$P = \frac{qwh}{550} = \frac{qh}{8.8} \tag{2}$$

From this equation it will be noted that *the cubic feet per second flowing in a stream multiplied by the available head and divided by 8.8 will give the total available horse power of the stream.* This is a convenient expression for power which should be kept in mind by the water power engineer.

Second: By the pressure of the water column on a given area exerted through a definite space. This method of utilization is represented by the equation

$$P = \frac{.434h \ as}{550t} \tag{3}$$

In this case .434h equals the pressure per square inch, and when multiplied by the area *a*, gives the total pressure in pounds exerted through the space *s*.

Third: By the momentum of the water exerted under the full velocity due to the head. The energy of a moving body in foot pounds per second is given by the formula

$$E = \frac{Wv^2}{2g} \tag{4}$$

The equation for the horse power of water under motion is therefore represented by the equation

$$P = \frac{qwv^2}{550 \times 2g} \tag{5}$$

An analysis of these formulas will show that under any given conditions the theoretical power exerted will be the same in each case.

17. **Natural Limit to Efficiency.**—The total energy in any working medium such as water, steam, air, etc., is the energy measured from the basis of the absolute zero for the medium which is being considered. For example, the average surface of Lake Michigan is 580 feet above sea level; each pound of water at lake level, therefore, contains 580 foot pounds of potential energy. This amount of energy must be expended in some manner by each pound of water passing from the lake level to the ocean level, which may be regarded as the absolute zero reference plane for water power. This energy cannot be utilized at Chicago for there no fall is available. A small portion of this energy is now utilized at Lockport, Illinois, from the Chicago Drainage Canal, where a fall of some thirty-four feet is available. Perhaps ultimately in its entire course one hundred and seventy feet of fall may be utilized by the waters of the drainage canal, in which case the absolute available energy of each pound of water cannot be greater than shown by the following equation:

$$\text{Available energy} = \frac{580 - 410}{580} = \frac{170}{580} = .2931, \text{ or } 29.31 \text{ per cent.}$$

This example shows therefore, the limits which natural conditions place on the proportion of energy which it is theoretically possible to utilize. For such losses the engineer is not accountable except for the selection of the best location and the best methods for utilizing such energy. The problem for his solution is, what amount of this available energy can be utilized by efficient machines and scientific methods.

18. **Practical Limits to Efficiency.**—The preceding equation is the equation for an ideally perfect machine. Of this available energy only a portion can be made actually available. In practice losses are met at every turn. Some energy will be lost in friction, as radiated heat, some in the slip by runners, or as leakage from defective joints. In many other ways the energy applied may be dissipated and lost. From this it follows:

The amount of energy which can be utilized can never be greater than the difference between the amount supplied to any given machine or mechanism, and the amount lost or consumed in such machines by friction, radiation or in other ways. Hence it follows that the efficiency of a given machine, or the percentage of energy avail-

able, or which can be obtained from the machine, can never be greater than the following:

$$\text{Efficiency} = \frac{E - (E' + E'' + E''' + E'''' \text{ etc.})}{E} \text{ in which}$$

E = total energy available.

E' E'' E''' etc. = the energy lost in friction and in various other ways, in the machine or system, and rejected in the exhaust from the same.

Every transmission or transformation of energy entails a loss, hence, starting with a given quantity of energy, it gradually disappears through the various losses involved in the mechanism or machines used. Other things being equal, the simpler the transmission or transformation, the greater the quantity of the original amount of energy that can be utilized.

The term efficiency as here applied represents always the ratio between the energy obtainable from the mechanism, machine, or combination of machines and the actual energy applied to it, or in other words the rates of energy output to energy input.

The efficiency of a turbine or water wheel is the ratio between the energy delivered at the wheel shaft to the energy in the water which enters the wheel.

The efficiency of a hydro-electric plant is the ratio between the energy in the electric current delivered at the switch board and the energy in the water entering the water wheel.

The efficiency of the dynamo in the same plant is the ratio between the energy furnished by the dynamo and the energy applied to it.

If a shaft receives from an engine 100 horse power and delivers ninety, ten horse power being lost in friction, etc., the efficiency of the shaft transmission is ninety per cent.

If a water wheel receives ten cubic feet of water per second under eighty-eight feet of head, the power input would be 100 horse power; and if it delivers eighty horse power at the shaft, its efficiency is eighty per cent.

If a steam engine receives 1,000,000 heat units from the steam it uses, and is able to deliver only the equivalent of 10,000 heat units; i. e., 7,780,000 foot pounds of work, the efficiency of the engine is only one per cent.

19. **Efficiency of Single Machines.**—The efficiency of any machine depends upon various factors, partially under control of the engineer and designer and partially uncontrollable. Friction of bearings,

atmospheric friction, leakage and slip involve losses of energy which may be reduced by first class design and construction, but with the possible exception of leakage, cannot be entirely obviated. Internal friction and various other losses depend both on the nature of the working medium (water, air, steam or electric current) and on

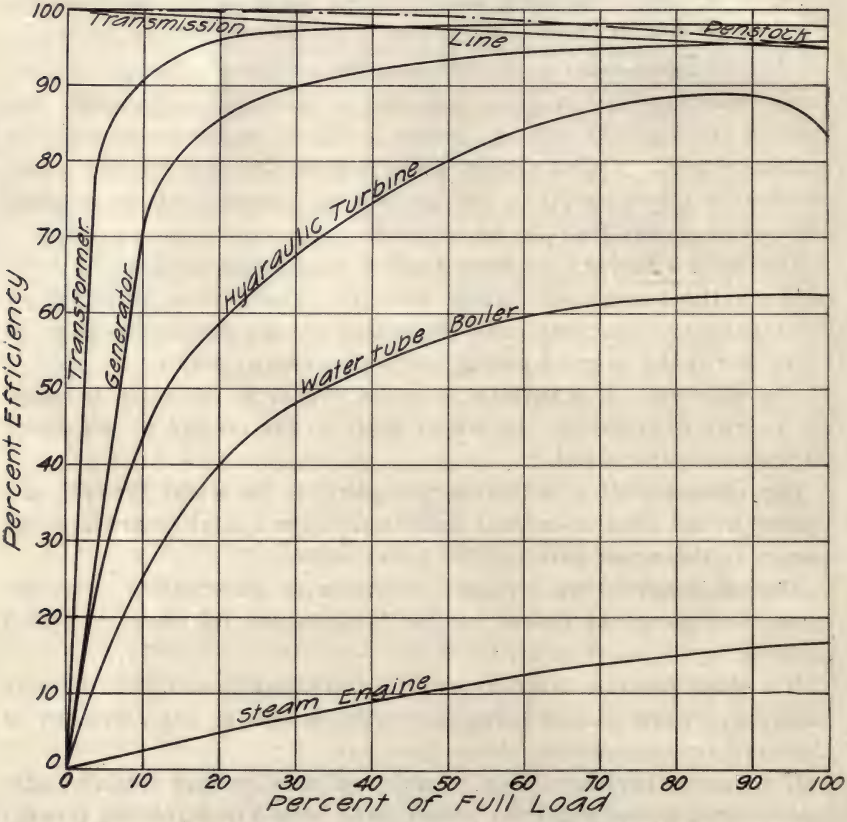

Fig. 19.—Efficiency Load Curves of Various Single Machines (see page 31).

the design, construction, maintenance and conditions of operation of the machine.

In all cases of units with moving parts, certain losses will always obtain when the machine is in operation, regardless of the load, while certain other losses will increase or vary with the load. In such cases, the best efficiencies of the machine are obtained under the condition of load during which the ratio of useful work done by the machine to the losses that obtain is the greatest. It is therefore obvious that it is usually desirable to so select, install, maintain and

operate machinery that it may work as nearly as possible with the least comparative losses or under the most efficient conditions.

With the varying conditions of power demand (see Chapter III) under which most plants operate, this ideal condition of operation can be only roughly approximated, but the maximum practical economy of the machine necessarily depends upon approximating this ideal condition as nearly as possible.

Figure 19, page 30, shows various efficiency curves of machinery commonly utilized in connection with power developments, and shows how the efficiency of such machines varies under varying conditions of load from no load to maximum load, and being usually at a maximum at or near full load. From similar diagrams of other machines which are to be used in any installation, can be seen and determined the general effect of any particular unit on a combined installation under the varying conditions of load. Occasionally, however, the efficiency curve of an individual machine is not a criterion of its true effect upon the economy of a combined plant. This occurs, for example, when the waste heat from a steam or other heat engine is used for heating purposes, and its effect on the economy of the installation is therefore more favorable than indicated by its efficiency-power diagram.

The selection of any machine or combination of machinery should involve a careful investigation of the probable efficiency load curves of the actual machine or machines which are to be utilized. Fig. 19 illustrates only typical efficiency load curves, from which individual machines must be expected to vary, often to a considerable extent.

20. Efficiency of a Combined Plant.—In any plant or connected arrangement of mechanisms and machines for the generation, transformation or transmission of energy, the efficiency of the plant is the product of the efficiency of each of its parts under the particular conditions of operation and load. The load conditions are seldom those of full load and are ordinarily those of part load.

Hence, to estimate total efficiencies, the efficiency of each unit of the system must be estimated under its actual load condition, and the combined efficiency can then be obtained. From the same calculation, the necessary relations between the input of energy into the system and the output of energy from the system can be obtained. Thus, if a turbine operating at half load has an efficiency of eighty per cent., and a pump direct-connected thereto has an efficiency of sixty per cent., the combined efficiency will be forty-eight per cent.; while if operated under full load, the efficiencies might

be eighty-five per cent. and seventy per cent. respectively, and the combined efficiency would reach 59.5 per cent.

To show in greater detail the various losses encountered in the generation and transmission of energy, especially as applied to hydro-electric plants, attention is called to Fig. 20, page 33. In this diagram is traced the losses under one condition of load from the potential energy of the water in the head race of the power plant to the power available at the point where it is used. In each case considered it is assumed that 1,000 horse-power of energy is applied to the particular work considered.

First, consider the transmission of power for traction purposes. If a certain head is available when no water is flowing in the raceways, that head becomes reduced at once when the wheels begin to operate. A certain amount of head is also lost in order to overcome the friction of flow through raceways, racks and gateways. In the problem here considered it is assumed that the above losses are five per cent. of the total energy available in the head-race, and that this loss occurs before the water reaches the turbines: hence, ninety-five per cent. of the potential energy is available at the turbine. The turbine loss is here assumed to be about twenty per cent. First-class turbines under three-quarter to full load conditions, will commonly give eighty per cent. efficiency, or a little better.

The next loss shown on the diagram is the loss in transmitting the energy through the bevel gear and the shafting to the generator. The loss in gearing, shafting, etc., is shown as ten per cent., which is probably much less than actually takes place in plants of this kind.

The loss in the transformation of power in the generator is given as eight per cent. The generator is an alternator, and the current generated might be at 2,300 volts. This current must be raised to a higher voltage, by means of transformers, for long distance transmission. These transformers might give an efficiency of about ninety-six per cent. The line loss is dependent on the size of the copper used, but would probably be designed for a loss not exceeding ten per cent. At the distributing point, where the energy is to be used, the high voltage current must be transformed again into suitable voltage for distribution. The same energy loss is estimated for these stepdown transformers. If the current is to be used for traction purposes, it will be necessary to convert it into direct current by means of a rotary converter, the efficiency of which is estimated at ninety-two per cent. The voltage from the general distribution system would probably be too high for direct use in the

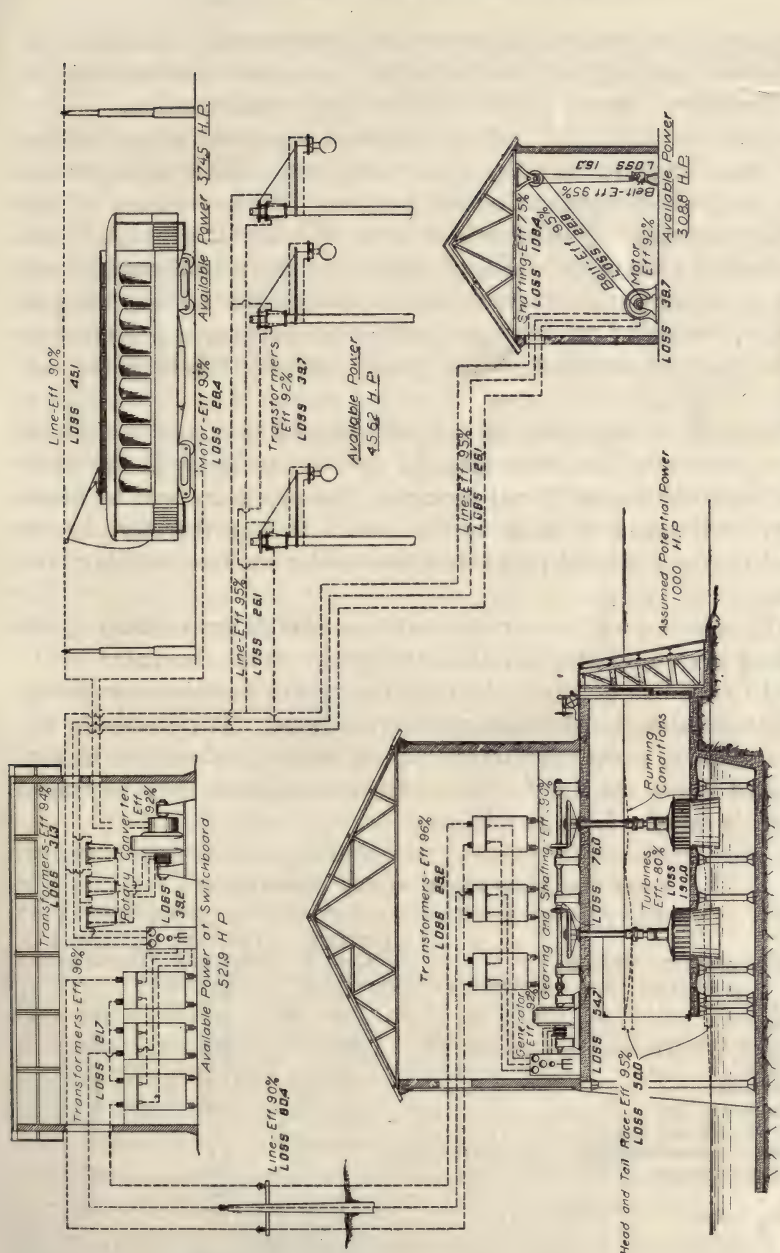

Fig. 20.—Diagram Showing Hydro-electric Power Plant With Power Applied to Various Uses and With Approximate Full Load Power Losses at Each Transformation (see page 32).

rotary converter, and would have to be transformed to a lower voltage before passing into the converter. A loss of about six per cent., therefore, should be allowed for this transformation.

The current from the rotary converter is subject to a line loss which may be again assumed at ten per cent. The loss in the car motor may be estimated at seven per cent. The percentage of loss and the percentage of efficiency for each unit in this generation and transmission system is based, of course, on the actual energy supplied by the unit next previous to it in the system, so that the percentages mentioned are not based on the total potential power available in the head-race but on the power actually reaching the machine.

In the solution of any actual problems of this character it is necessary to determine the efficiencies of the various units of the plant under the condition of actual service. The efficiency will be found to vary under various conditions of load. It may therefore be desirable to determine the probable losses under various working conditions.

In the selection of the various machines which are to form a part of such a system of transmission, the choice should be based on an effort to establish a plant which will give the maximum economy when all conditions of loading are considered. The losses in the transmission of power for traction purposes, as shown on the diagram and under the fixed load conditions assumed, may be traced through in tabular form as follows:

Total Energy Available = 1000 Horse Power

Mechanism or Machine	Per Cent Loss	Per Cent Efficiency	Loss in Horse power
Head race	5	95	50
Turbine	20	80	190
Shaft and gearing	10	90	76
Generator	8	92	54.7
Transformers	4	96	25.2
Transmission line	10	90	60.4
Step-down transformers	4	96	21.7
Secondary transformers	6	94	31.3
Rotary converters	8	92	39.3
Line	10	90	45.1
Traction motor	7	93	28.4

Total power utilized for operating the cars, 374.5 Horse Power or 37½ per cent. of the original energy.

In the generation and transmission of power for lighting purposes, the losses will be similar to those above mentioned, up to and including the step-down transformers at the point of distribution. In this case, however, no secondary transformers or rotary converters would be necessary. The only loss between the step-down transformers and the light will be the line loss assumed at five per cent. The loss in the individual transformer for the light will be about eight per cent., leaving the available energy for actual use in the lamp at about 456.2 horse power, or a little less than forty-six per cent. of the total energy in the head-race.

In the case of the utilization of this energy for manufacturing purposes, the loss would be the same up to and including the step-down transformers at the point of distribution. The line loss in the distribution from the transformer house to the manufacturing establishment may be assumed at five per cent. The motor, if properly selected, may be run at the line voltage, and no transformer losses need be considered. The motor efficiency is here shown at ninety-two per cent., although in most cases the percentage of efficiency would be considerably less.

The belt loss in transmitting the power from the motor to the line shafting is estimated at five per cent.

The shafting necessary for the general distribution of power through the factory is estimated at seventy-five per cent. efficiency.

The belt loss from the shaft to the individual machine is estimated at an additional five per cent., leaving the total energy available for use in the machine at 308.8 horse power, or about thirty-one per cent. of the original energy in the head-race.

It should be noted that in each of the three transmission systems mentioned above, the actual power utilized at the point of application is less than half of the energy available in the head-race. It is the function of the engineer to see that these losses are reduced to the greatest practicable extent. These losses must be limited in both directions. They must not be too great, nor too small. They must be adjusted at the point where true economy would dictate. This limit is the point where the capitalized value of the annual power lost is equal to the capitalized cost of effecting further saving. In other words, true economy means the construction of a plant that will save all the power or energy which it is financially desirable to save, and will permit such waste of energy as true economy directs.

21. Average Working Efficiencies of an Actual Plant.—That losses similar to those estimated in the plant previously described are actually incurred in hydro-electric plants is shown by the records of the Seattle Municipal Light and Power Plant.

Table 3, page 37, gives an outline of the losses and efficiencies in this plant for 1911. These data were presented by J. D. Ross in a paper before the Pacific Coast Section of the American Institute of Electrical Engineers (see Trans. Am. Inst. E. E. Vol. XXXI, p. 471). The figures given are believed to approximate closely the true values, since great care was taken in the measurements which were made with frequently calibrated instruments. All results were checked in as many ways as practicable.

The Seattle plant is a hydro-electric system delivering water to two 1500 KW Pelton units and two 5000 KW turbine units under 600 feet head through two pipes approximately three and one-half miles long, one of which is sixty-seven and three-fourths and the other forty-nine inches inside diameter. The current is transmitted at 60,000 volts through two lines to Seattle, a distance of thirty-eight and seven-tenths miles, and is there distributed at 15,000 and 2,400 volts for use by approximately 20,000 customers and for the city street lighting.

22. Graphical Calculation of Plant Losses at All Loads.—Fig. 21, page 38, shows a graphical analysis of the losses which might be entailed and the efficiencies which might be secured in each unit and in a combined modern hydro-electric plant of first class design and construction, and under all conditions of load from full load to no load. The method used in making this graphical analysis is fairly self-explanatory and is discussed in greater detail by the author in another volume (see "Hydraulic Machinery"). The last efficiency curve on the right of this diagram shows the plant efficiency and is based on the ratio of energy delivered at the customer's switchboard to the energy of the water input to the hydraulic plant. The average resulting efficiencies of a plant for the day, week or year will depend upon the varying load which the plant must carry; it will never equal the full load efficiency of the plant input except under unusual or test conditions. From this diagram it will be noted that a first class plant should deliver from sixty-five per cent. to sixty-eight per cent. of the theoretical power of the water even with considerable fluctuation in load, provided the load is not too small in comparison to the plant capacity.

TABLE 3.

Outline of Losses and Efficiencies for 1911, Seattle Municipal Light and Power Plant.

	Per Cent All Day Efficiency	Total 1911 Input KWH	Average 1911 Input KW	Total 1911 Loss KWH	Average 1911 Loss KW	Per Cent Loss	Pct. of Penstock Input	Pct. of Total Loss
Generating system	54.4	52,639,000	6,009	23,990,300	2,739	45.6	45.6	75.3
Penstocks	97.7	52,639,000	6,009	1,214,900	139	2.3	2.3	3.8
Generating station	55.7	51,424,100	5,870	22,775,400	2,600	44.3	43.2	71.5
Water wheels	60.7	50,758,900	5,795	19,944,400	2,277	30.3	37.9	62.6
Generators	93.5	30,814,500	3,518	1,990,800	227	6.5	3.8	6.2
Exciters	....	665,200	76	665,200	76	....	1.3	2.1
Station lights and control	....	175,000	20	175,000	20	....	0.3	0.5
Transmission system	91.6	28,648,700	3,270	2,413,500	276	8.4	4.6	7.6
Step-up transformers	96.1	28,648,700	3,270	1,126,000	129	3.9	2.1	3.5
Transmission lines	98.6	27,522,700	3,141	378,000	43	1.4	0.7	1.2
Step-down transformers	96.6	27,144,700	3,098	909,500	104	3.4	1.7	2.9
Distributing system	79.2	26,235,200	2,994	5,448,700	622	20.8	10.3	17.1
City sub-station	98.7	26,235,200	2,994	346,400	40	1.3	0.7	1.1
Station lights and control	....	317,400	37	317,400	37	1.2	0.6	1.0
Switchboard meters	....	....	....	29,000	3	....	0.1	0.1
15,000 volt system	92.5	11,587,000	1,323	868,600	90	7.5	1.6	2.7
15,000 volt lines	99.2	11,587,000	1,323	93,500	11	0.8	0.2	0.3
15,000 volt transformers	93.2	11,493,500	1,312	775,100	88	6.8	1.5	2.4
Series street lights	86.3	2,672,800	305	367,200	42	13.7	0.7	1.2
Transformers	95.0	2,672,800	305	133,700	15	5.0	0.3	0.4
Series circuits	90.8	2,539,100	290	233,500	27	9.2	0.4	0.7
Cluster street lights	79.1	1,486,000	170	310,000	35	20.9	0.6	1.0
Cluster transformers	87.8	1,486,000	170	181,000	21	12.2	0.3	0.6
Underground cables	90.1	1,305,000	149	129,000	15	9.9	0.2	0.4
2,400 v. commercial system	76.2	13,178,400	1,612	3,123,700	357	23.8	5.9	9.8
Feeder regulators	98.6	13,178,400	1,612	178,500	20	1.4	0.3	0.6
Primary feeders	96.0	12,999,900	1,592	521,600	60	4.0	1.0	1.6
Transformers	88.8	12,478,300	1,532	1,391,000	159	11.2	2.6	4.4
Secondaries	92.9	11,087,300	1,373	782,600	89	7.1	1.5	2.5
Customers' meters	97.6	10,304,700	1,284	250,000	29	2.4	0.5	0.8
Direct-current system	35.7	673,200	77	432,800	49	64.2	0.8	1.4
Motor-generator	38.0	673,200	77	417,000	48	62.0	0.8	1.3
D. C. circuits	95.0	256,200	29	12,800	1	5.0	....	....
Customers' meters	98.8	243,400	28	3,000	....	1.2	....	....

Total power loss............................ 31,852,500 KWH Av. 3,636 KW
Total power delivered to customers....... 17,304,900 KWH Av. 1,975 KW
Total power delivered to street lamps.... 3,481,600 KWH Av. 398 KW
Total delivered power...................... 20,786,500 KWH Av. 2,373 KW

Over-all efficiency, 39.5 per cent.
(1 KWH at the customers' premises requires 1.364 gals.—5.163 of water from Cedar Lake at average head of 590 ft.—179.8 m.)

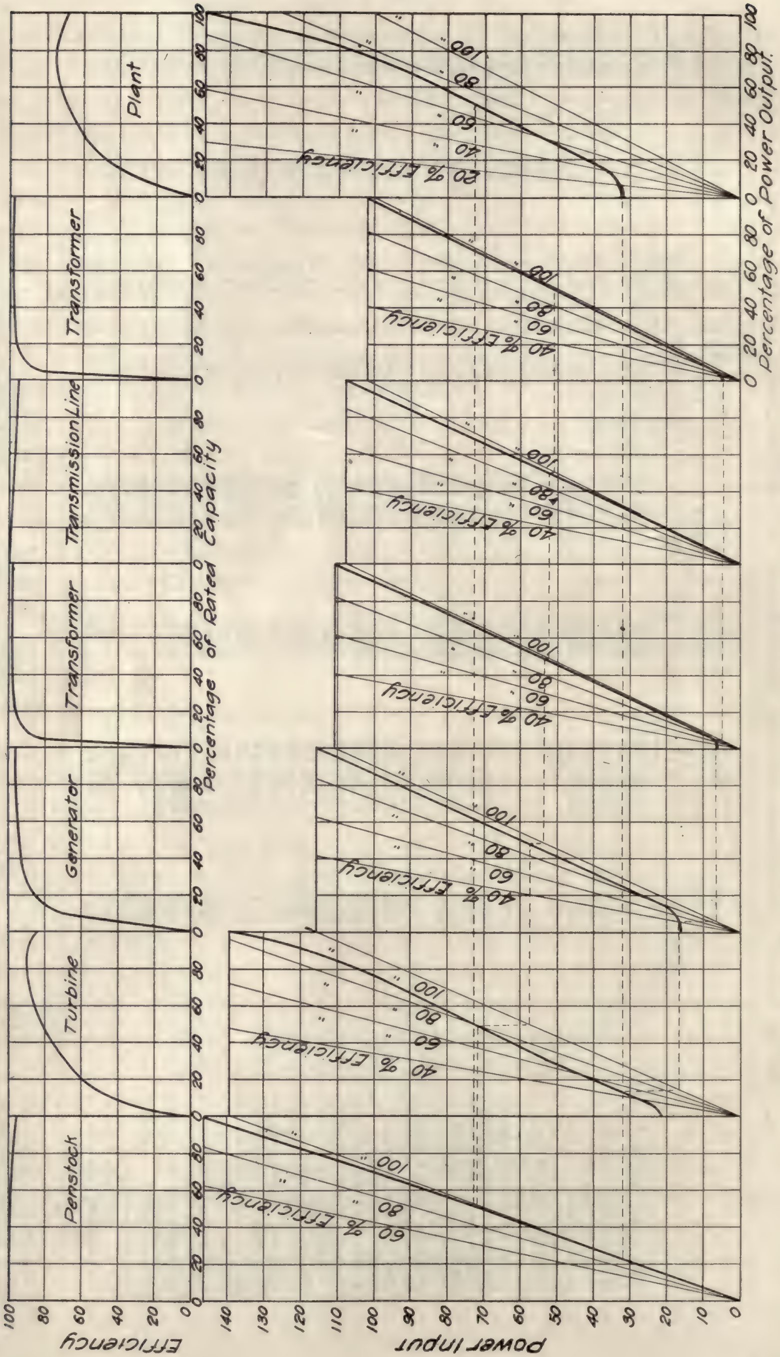

Fig. 21.—Graphical Analysis of Plant Losses (see page 36).

It should be noted, however, that poor design or construction frequently reduces this output to fifty per cent. or less of the theoretical power of the stream.

23. **Units of Energy.**—Energy is known by many names and exists in many forms which seem more or less independent. The principal forms of energy are measured by various units (see Table 5, page 42). Those most commonly considered in power development and transmission are as follows:

Work is energy applied to particular purposes. In general it is energy overcoming resistance, mechanically it is the exertion of force through space.

Power is the rate of work, or the relative amount of work done in a given space of time.

The unit of work is the foot pound, or the amount of work required to raise one pound one foot. One pound raised one foot, one-tenth pound raised ten feet, ten pounds raised one-tenth of a foot, or any other sub-division of pounds and feet whose product will equal unity requires one foot-pound of work to perform it.

The unit of power is based on the unit of work, and is called "horse power." It is work performed at the rate of 550 foot pounds per second, or 33,000 foot pounds per minute.

Units of Heat. The unit of heat is the amount of heat which will raise one pound of water from thirty-nine degrees Fahr. to forty degrees Fahr. at atmospheric pressure. It is called the British Thermal Unit, and is indicated by the initials B. T. U.

Electric Unit. The unit of quantity of electricity is the coulomb. One coulomb per second is called an ampere, and one ampere under a volt pressure is equal to a watt, the unit of electric power.

Water Power. Water power is the power obtained from a weight of water moving through a certain space. In water power the unit of quantity may be the gallon or the cubic foot; the unit of head may be the foot; and the unit of time may be the second or minute. The weight of water, unless highly mineralized, at ordinary temperature, varies from 62.3 to 62.5 pounds per cubic foot. As these weights vary from each other less than one-third of one per cent., the difference is insignificant in practical problems where the errors and uncertainties are often large. In the further discussion of this subject, therefore, the weight of 62.5 pounds is used as the most convenient in calculation (see Table 5, page 42).

Steam Power. The unit of steam power in ordinary use is the pound of steam, its pressure, and rate of use. It is, however, based

on the heat unit, and must be so considered for detailed examination.

Definite quantities of work are also designated by the "horse power hour" equivalent to 1,980,000 foot pounds, and the "kilowatt hour," equivalent to 2,654,150 foot pounds.

The pound of steam may be considered as containing an average of 1,000 British thermal units, which may be utilized for power. This is equivalent to 778,000 foot pounds.

24. Conversion of Energy Units.—The various forms of energy as expressed by the units named are convertible one into another in certain definite ratios which have been determined by the most careful laboratory methods. In considering these ratios, however, it must be remembered that, as shown in the preceding examples, in the transformation from one form of energy into another the ratios given cannot be attained in practice on account of losses which can not be practically obviated. Such losses must be, in good practice, reduced to a minimum, and the ratios given are, therefore, the end or aim toward which good practice strives to attain as nearly as practicable when all conditions and facts are duly considered.

Energy must be considered in two conditions as well as in the above named forms, viz.: passive and active or potential and kinetic.

Potential energy is energy stored and does not necessarily involve the idea of work. Kinetic energy is energy in action and involves the idea of work done or power exerted and for its measurement must be considered in relation to time.

The most common units of potential energy and their equivalents are as follows (see also Table 4, page 42) those most commonly used being printed in italic type:

The foot pound (one pound raised one foot)
- =1/62.5 or .016 foot cubic foot (of water).
- =1/8.34 or .12 foot gallon (of water).
- =1/2655.4 or .0003766 volt coulombs.
- =1/778 or .001285 British thermal units.

The foot cubic foot (one cubic foot of water raised one foot)
- =*62.5 foot pounds.*
- =*7.48 foot gallons.*
- =.08 British thermal units.
- =.02353 volt coulombs.

The foot gallon (one gallon of water raised one foot)
- =*8.34 foot pounds.*
- =.01072 British thermal units.
- =.00314 volt coulombs.
- =.1334 foot cubic feet.

The volt coulomb
= 2655.4 foot pounds.
= 42.486 foot cubic feet.
= 318.39 foot gallons.
= 3.414 British thermal units.

The British thermal unit
= *778 foot pounds.*
= 12.448 foot cubic feet.
= 93.28 foot gallons.
= .2929 volt coulombs.

Quantities of energy available, used or to be used, and either potential or kinetic may be measured in the above units.

When the rate of expenditure is also stated these units express units of power. Some of the equivalent values of power are as follows:

The horse power
= *1980000 foot pounds per hour.*
= *33000 foot pounds per minute.*
= *550 foot pounds per second.*
= 31680 foot cubic feet per hour.
= *528 foot cubic feet per minute.*
= *8.8 foot cubic feet per second.*
= 237600 foot gallons per hour.
= *3960 foot gallons per minute.*
= 66 foot gallons per second.
= *746 watts.*
= 2545 British thermal units per hour.
= 42.41 British thermal units per minute.
= .707 British thermal units per second.

The foot pound per minute
= 1/33000 or .0000303 horse power.
= 1/778 or .00129 British thermal units per minute.
= .0226 watts.
= 1/8.34 = .12 foot gallons per minute.
= 1/62.5 = .016 foot cubic feet per second.

The foot cubic foot per minute
= *62.5 foot lbs. per minute.*
= 1/528 = .00189 horse power.
= 1.412 watts.
= 7.48 foot gallons per minute.
= .0803 British thermal units per minute.

The foot cubic foot per second
= 3750 foot lbs. per minute.
= 62.5 foot lbs. per second.
= 1/8.8 = .1136 horse power.
= 448.8 foot gallons per minute.

= 7.48 foot gallons per second.
= 4.820 British thermal units per minute.
= .0803 British thermal units per second.

The watt

= 44.24 ft. lbs. per minute.
= .00134 horse power.
= .0568 British thermal units per minute.
= 5.308 foot gallons per minute.
= .7089 ft. cu. ft. per minute.

The British thermal units per minute

= *778 ft. lbs. per minute.*
= .02357 horse power.
= 17.58 watts.
= 93.28 ft. gal. per minute.
= 12.48 ft. cu. ft. per minute.

TABLE 4.

Equivalent Units of Energy.

WORK				HEAT		ELEC-TRIC	HYDRAULIC			
Foot Pound	Foot Ton 2240 Lbs.	Kilogram Meter	Tonne Meter	B. T. U. Deg. Fah. Pound	Calorie Deg. Cent. Kilogram	Volt Coulomb	Foot Gallon	Foot Cubic Foot	Pound Gallon	Pound Cu. Ft.
1	.000446	1383	.000138	.001285	.000324	.000337	.12	.016	.0519	.0069
2240	1	309.688	.3097	2.8785	.7262	.8439	268.817	35.906	116.414	15.456
7.233	.00323	1	.001	.0093	.00235	.00272	.8673	.1159	.3755	.0499
7233.18	3.2291	1000	1	9.302	2.3452	2.7241	867.303	115.928	375.516	49.90
778	.3474	107.562	.1076	1	.2520	.2929	93.28	12.448	40.394	5.368
3085.34	1.3774	426.394	.4264	3.9683	1	1.1623	370.17	49.396	160.29	21.221
2655.4	1.1854	371.123	.3671	3.414	.8603	1	318.29	42.486	137.87	183.23
8.341	.00372	1.1532	.00115	.1072	.0027	.00314	1	.1334	.433	.05754
62.39	.02785	8.6257	.00863	.0803	.00202	.02353	7.48	1	3.245	.4312
19.259	.00859	2.6626	.00266	.0248	.00624	.00726	2.309	.3082	1	.1329
144.92	.0647	20.036	.02004	.1863	.04712	.05457	17.37	2.318	7.524	1

TABLE 5.

Equivalent Measures and Weights of Water at 4° Centigrade—39.2° Fahrenheit.

U. S. Gallons	Imperial Gallons	Liters	Cubic Meters	Pounds	Cubic Feet	Cubic Inches	Circular Inch 1 Foot Long
1	.83321	3.7853	.0037853	8.34112	.13368	231	24.5096
1.20017	1	4.54303	.004543	10.0108	.160439	277.274	29.4116
.264179	.22012	1	.001	2.20355	.035316	61.0254	6.4754
264.179	220.117	1000	1	2203.55	35.31563	61025.4	6475.44
.119888	.099892	.453813	.0004538	1	.0160266	27.694	2.9411
7.48055	6.23287	28.3161	.0283161	62.3961	1	1728	183.346
.004329	.003607	.0163866	.0000164	.0361089	.0005787	1	.10613
.0408	.034	.1544306	.0001544	.340008	.005454	9.4224	1

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CHAPTER III

THE LOAD

THE LOAD CURVE AND LOAD FACTOR, AND THEIR INFLUENCE ON THE EFFICIENCY AND DESIGN OF WATER POWER PLANTS

25. **Variation in Load.**—All power plants are subjected to more or less change in load, and this continually changing load has an important bearing on the economy of the plant, and should be carefully considered in its design and construction.

If the power output of any plant be ascertained, minute by minute or hour by hour, either by means of recording devices or by reading the various forms of power indicators usually provided for such purposes, and a graphical record of such readings be made, a curve varying in height, in proportion as the power varies from time to time, will result. This curve is termed the daily load curve. The load curve itself will vary from day to day as the various demands for power vary, but it usually possesses certain characteristic features which depend on the load tributary to each plant and which vary somewhat as the seasons or other conditions cause the load to vary.

The characteristics of the load curve, due to certain demands, can be quite safely predicted. A power plant in a large city, for example, will carry a comparatively small continuous night load. This, in dark weather and in winter, will be increased by the early risers who are obliged to go early to shop and factory. These demands usually begin to affect the load curve about 5 A. M. and may cease wholly, or in part, by 7 A. M., depending on the season and latitude. From 7 to 8 A. M. the motor load begins to be felt. This may reach a maximum from 10 to 12, and usually decreases from 12 to 2 during the lunch hours. The maximum load usually comes in the afternoon when business reaches a maximum, and when the largest amount of power and also light (in the late afternoon) are used. The load begins to decrease after the evening meal, as the demand for light lessens, and may again increase somewhat as the theatres and halls open for evening's amusements. The character of the load curves, due to various loads, is best understood by a study of the actual curves themselves.

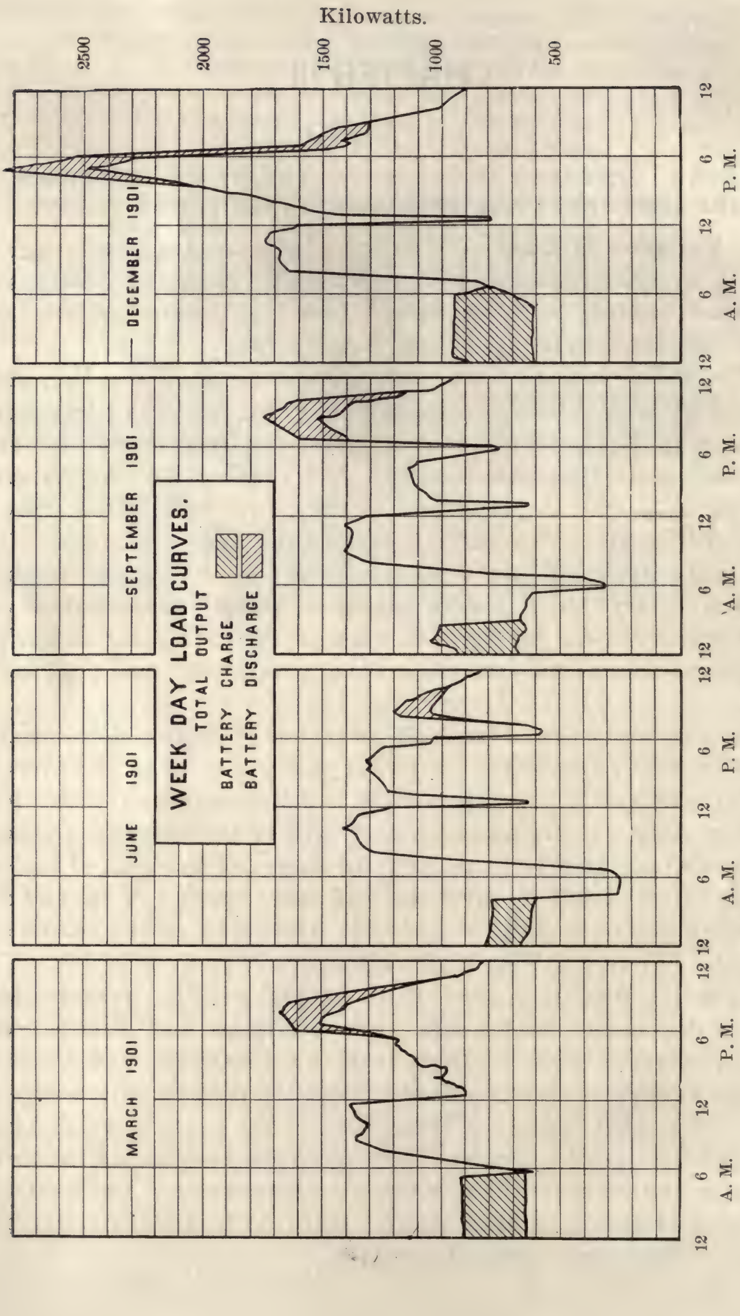


Fig. 22.—Hartford Electric Light Co., Load Curves (see page 47).

26. Load Curves of Light and Power Plants.—The curves shown in Fig. 22, page 46, are from the plants of the Hartford Electric Light Co., of Hartford, Conn., and will illustrate variation of the load curve at different seasons of the year (see "The Electrical World and Engineer" March 8th, 1902). This plant is a combined water and steam power plant, and is provided with a storage battery to assist in equalizing the load. These curves are described as follows:

"On a week day in January, 1901, the maximum load was 2720 k. w. and the total energy output was 30249 k. w. hours. The average hourly load was then 1260 k. w. or forty-six per cent. of the maximum load. On this same day the battery discharged at the rate of 260 k. w. at the peak of the load. In the early morning hours of this day the load on the system, apart from battery charging, reached its minimum at 612 k. w., or only 22.5 per cent. of the maximum load. In July, 1901, the maximum load on a certain week day was 1390 k. w., and the minimum 250 k. w., or eighteen per cent. of the former. The total output on this day was 25,105 k. w. hours, so that the average load during the twenty-four hours was 1046 k. w. or seventy-five per cent. of the maximum. In January, the maximum load came on between 4 and 5 P. M., when lighting was the predominant factor, but in July the greatest demand came on the system in the latter part of the forenoon, and must have been made up in large part by requirements for electric power. By December 1901, the maximum load reached 2838 k. w. and the minimum 612 k. w. The approximate capacity of all connected lamps and motors in that month was 8530 k. w. The maximum load for the December day of 2838 k. w. is only thirty-three per cent. of the connected capacity. On this day the total output was 32,191 k. w. hours, so that the average load during the twenty-four hours was 1342 k. w. This average is fifteen per cent. of the total capacity of connected lamps and motors."

Figure 23, page 48, shows daily load curves from the Christiania Power Stations, of Christiania, Norway. In this figure are shown the maximum, the minimum, and a mean curve for the entire year. The difference between the maximum and minimum curves is here very marked. This is readily ascribed to the high latitude of Christiania as the long twilights of summer render lighting at that season almost unnecessary, while the very short and dark days of

winter create not only a high maximum but a high continual demand during the entire day. No data as to kind of load is available.

Figure 24, page 49, is a power curve from the New York Edison Company. On August 1, 1905, there were connected up to the system of the New York Edison Company an equivalent of 1,651,917 incandescent lamps, 22,093 arc lamps, 2,539 k. w. in storage batteries and

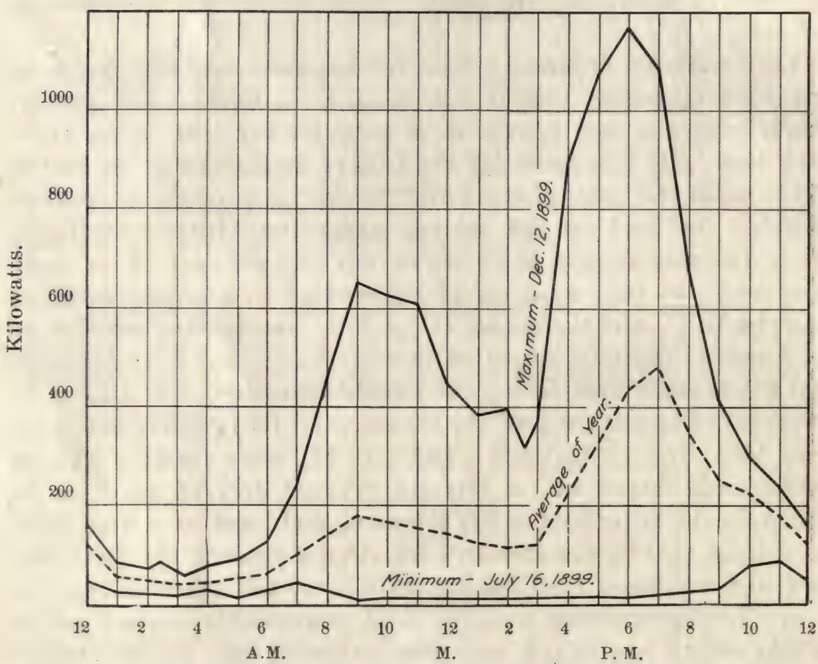
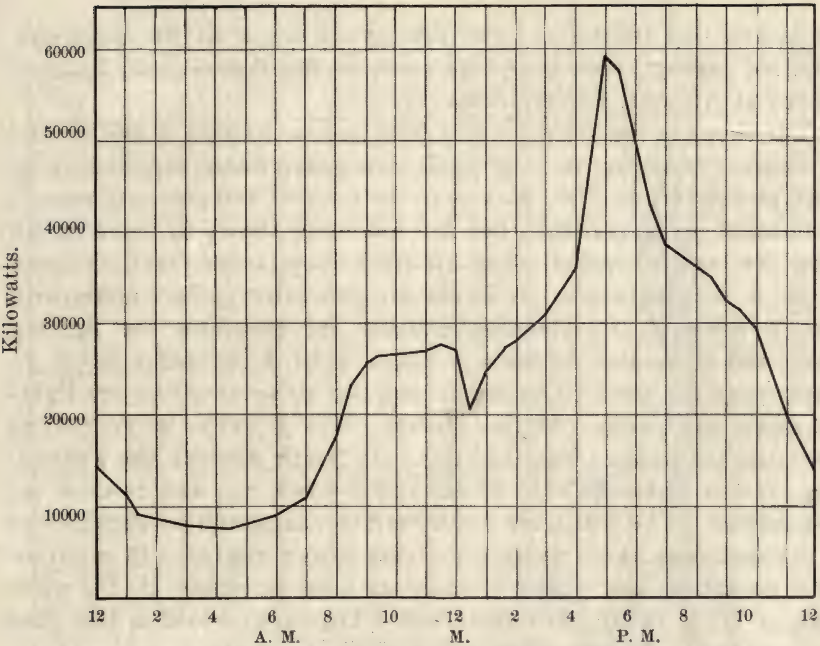


Fig. 23.—Typical Electric Lighting Load Curves. Christiania, Norway, Power Stations (see page 47).

99,258 H. P. in motors. The lighting load forms 52.2 per cent. of the connected load.

The effect of extraordinary conditions on the load curve and the necessity of some kind of storage to provide for the same, is well illustrated by Fig. 25, page 49, which shows the effect on the load curve of a lighting plant of a sudden thunderstorm. When such a storm occurs in the late afternoon the light load from schools, offices, stores, etc., may be suddenly thrown on, and the result may be an extraordinary load which the plant must meet.

27. Factory Load Curves.—Shop and factory loads are supposed to be the most uniform in character, yet they are subject to great



New York Edison Co., Load Curve, Day of Max. Load, Dec. 22, 1904.

Fig. 24.—Typical Electric Lighting Load Curve (see page 48).

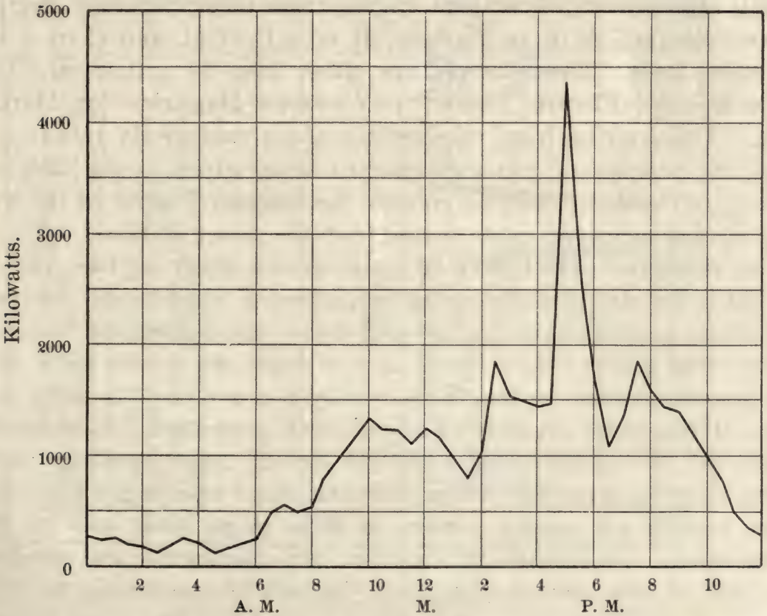


Fig. 25.—Sharp Thunder Storm Peak, Dickenson St. Station, Manchester, Eng. (see page 48).

variation, due to the sudden turning on or off of the machines. Fig. 26, page 51, shows the load curve of the Pennsylvania Railroad Shops at Altoona, Pennsylvania.

The shops of the Pennsylvania Railroad are located in and around Altoona, Pennsylvania, in groups, each group being supplied by its own power station. No data as to the number and power of motors connected up is available, but the following shows to some extent how the load is divided. The Machine Shop power plant embraces 3–300 k. w. generators, one Brush arc generator (power unknown), and a forty H. P. Thomson-Houston arc generator for lighting shop and grounds. At the Car Shops 4–250 k. w. and 1–625 k. w. generators are used. Current is supplied to seventy-five arc lights in shops and yards. At the Juniata shops 3–300 k. w. generators are used for power purposes only. At South Altoona the generating station embraces 1–50 k. w., and 2–500 k. w., and 2–300 k. w. generators. The loads are quite variable, as would be expected in a railroad shop, there being some very heavy machines in intermittent operation, one planer running as high as eighty H. P., while twenty H. P. motors are numerous. The normal load is less than the maximum, but the latter is frequently reached.

A, B and C, Fig. 27, page 52, are three typical factory load curves which represent types of load curves from three different electric power stations, A in an Eastern, B in a Central, and C in a far Western state. These curves are taken from an article on "The Economics of Electric Power" in Cassier's Magazine for March, 1894. The circuits from these stations are exclusively motor circuits, the number of motors connected being given in the table on page 53. On the circuits covered by the diagram B some of the motors are five miles and more distant from the power stations.

One deduction which may be made from a study of these curves is that in an electrical power system where a considerable number of motors are employed the initial dynamo plant need not be equal to the total motor load. In the case in hand the curves show that the generator need be but from twenty-five per cent. to forty per cent. of the rated capacity of the motors connected. In order to check off this phenomenal condition actual meter readings were taken monthly from fifty-three different shops covering a period of from four to six months, current to these shops being sold on the meter basis. The results showed that only twenty-five and one-half per cent. of the nominal capacity of the motors was employed, thus

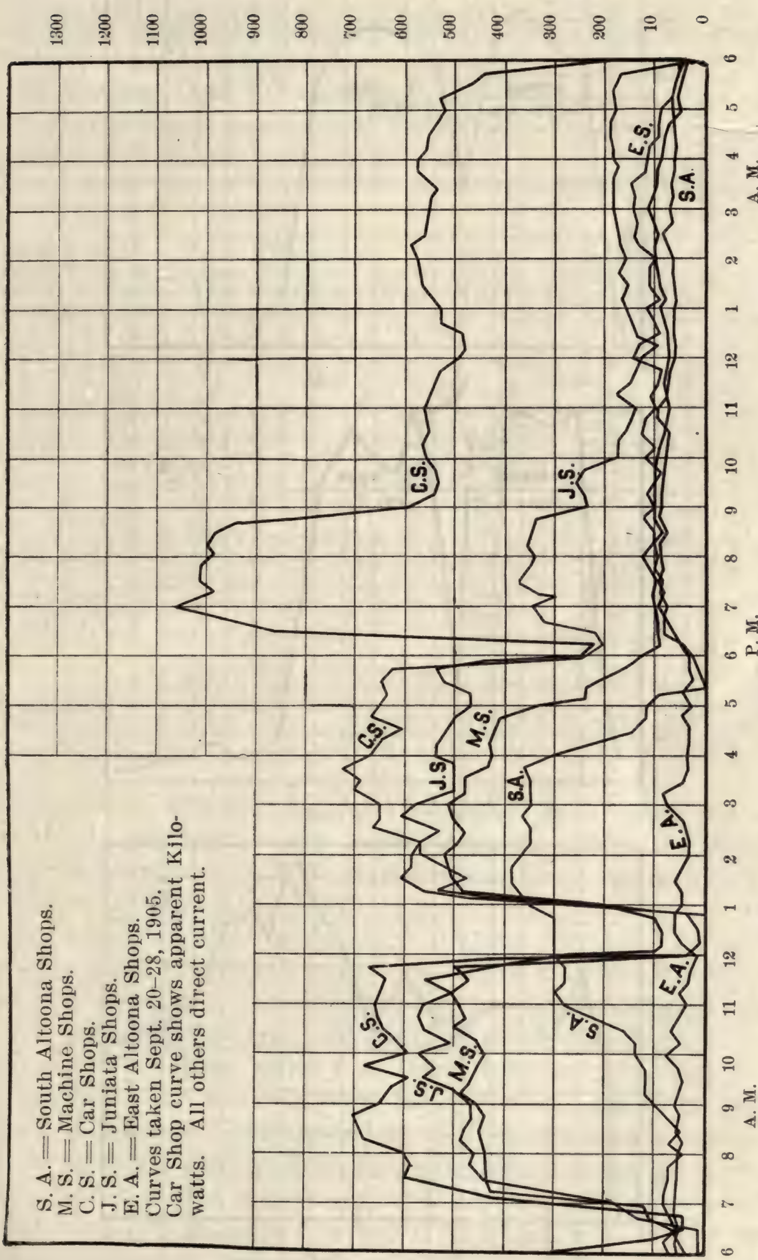


Fig. 26.—Load Curves Pennsylvania, R. R. Shops, Altoona, Pa. (see page 50). (From Electrical World and Engineer, Aug. 18, 1906).

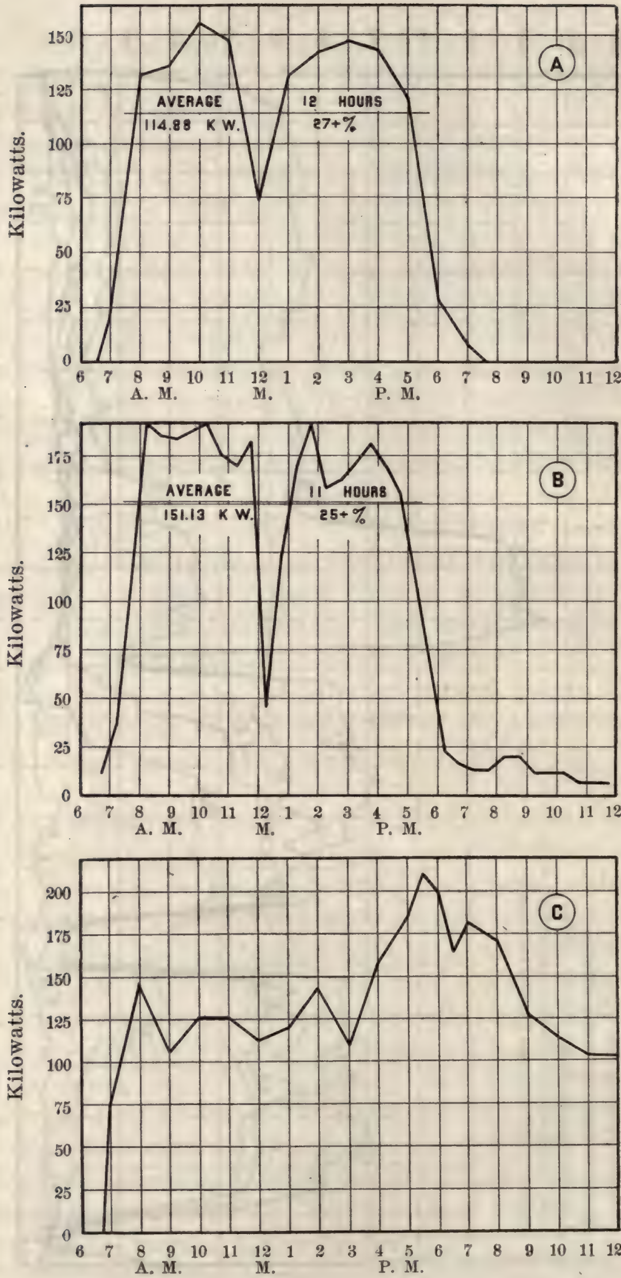


Fig. 27.—Typical Factory Load Curves (see page 50). (Cassier's Magazine.)

practically checking the conditions indicated by the diagrams of the central power stations.

28. Railway Load Curves.—The power load most subject to violent fluctuations is that utilized for railway purposes. The sudden

A			B			C		
Size of Motor (H.P.)	No. in Use	Combined H.P.	Size of Motor (H.P.)	No. in Use	Combined H.P.	Size of Motor (H.P.)	No. in Use	Combined H.P.
½	3	1½	⅛	3	⅜	¼	4	1
1	31	31	¼	2	½	½	1	½
2	10	20	½	1	½	1	5	5
3	19	57	1	15	15	2	3	6
5	10	50	2	14	28	3	4	12
7½	3	22½	3	5	15	5	3	15
10	12	120	5	12	60	6	5	30
13	5	75	7½	12	90	8½	3	25½
20	2	40	10	15	150	10	6	60
25	4	100	15	9	135	14	6	84
50	1	50	20	5	100	15	1	15
....	....	....	25	3	75	17½	1	17½
....	....	....	30	3	90	25	1	25
....	....	....	40	1	40	30	3	90
....	....	....	....	....	....	40	1	40
....	....	....	....	....	....	60	2	120
....	....	....	....	....	....	70	1	70
Total..	100	567	....	100	799⅜	....	50	616½

changes in the demand for power occasioned by stopping and starting of cars, which may, under some conditions, occur simultaneously are often very rapid and the resulting load fluctuations very great.

Figures 28 and 29, page 54, show two sets of curves taken from the power charts of the International Railway Company of Buffalo, which may be considered typical for electric railways. Each chart has two sets of curves, one for the city lines, on which the traffic is purely urban in character, and the other for the Tonawanda, Lockport and Olcott Line, which is an interurban line. In either set the total load at any time is represented by the ordinate to the highest curve in that set. The amount of load carried by any portion of the system is represented by the difference between the ordinates to the curve of that portion and to the curve next below. On the urban lines two peaks will be observed, one at 8 A. M. and one at 6 P. M., for both winter and summer, the afternoon peak of the former being nearly seventy-five per cent. greater than the latter,

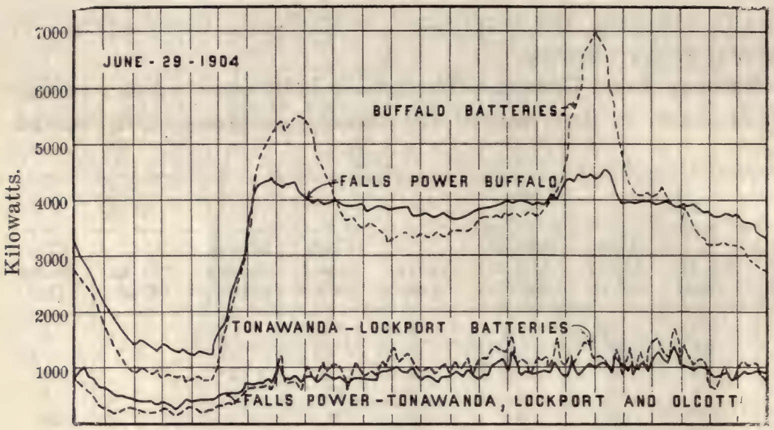


Fig. 28.

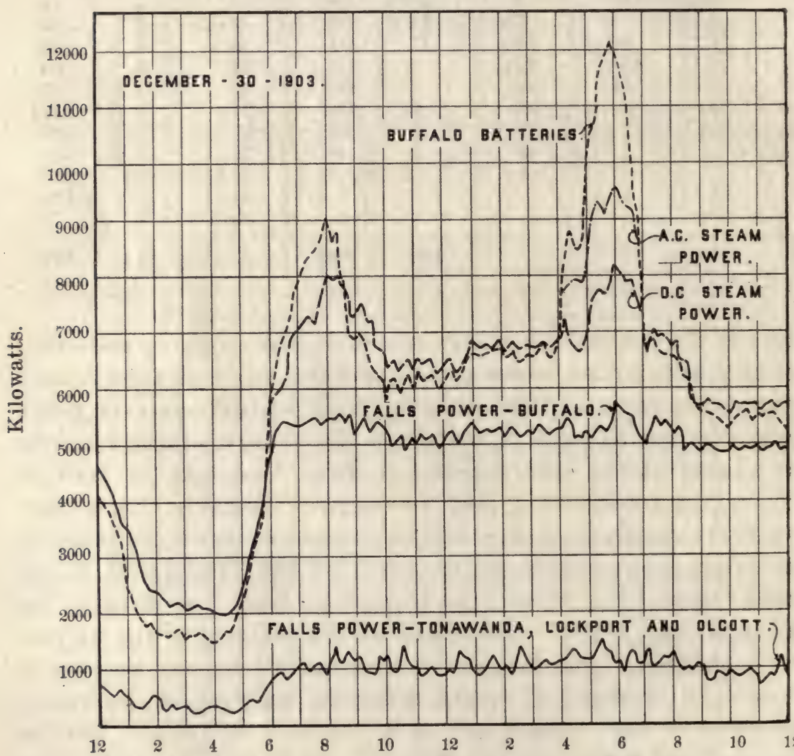


Fig. 29.—Typical Railway Load Curves, International Ry. Co. (from Electrical World and Engineer). (See page 53.)

however. The load curve of the interurban line appears to be nearly uniform throughout the year.

29. Growth of Load.—While occasionally a power plant may be designed for certain fixed conditions of load from which no material changes need be contemplated, in most cases such plants must be so installed as to provide for an ultimate increase with the normal

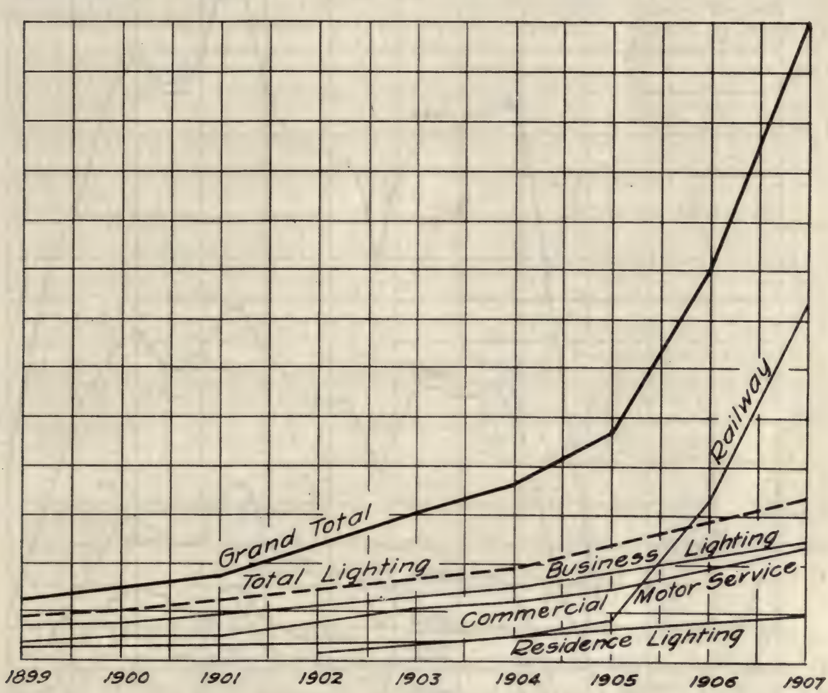


Fig. 30.—Growth of the Output of Commonwealth Edison Company of Chicago (see page 56).

growth of business. In the case of public utilities, a power plant must grow and develop both with the growth of the demand for energy which must be normally expected as the advantage of the utility becomes recognized by the community, and with the growth of population of the community as well.

The immediate demands for power on the completion of a plant will therefore seldom fix the ultimate limits for which a plant should be designed, and the probable future growth as well as the character of the immediate load, must be carefully considered in the selection of machinery and the provision for the future.

Figure 30, page 55, shows the comparative growth in the power demand for various services as well as the growth of total demand on the plants of the Commonwealth Edison Company of Chicago (see

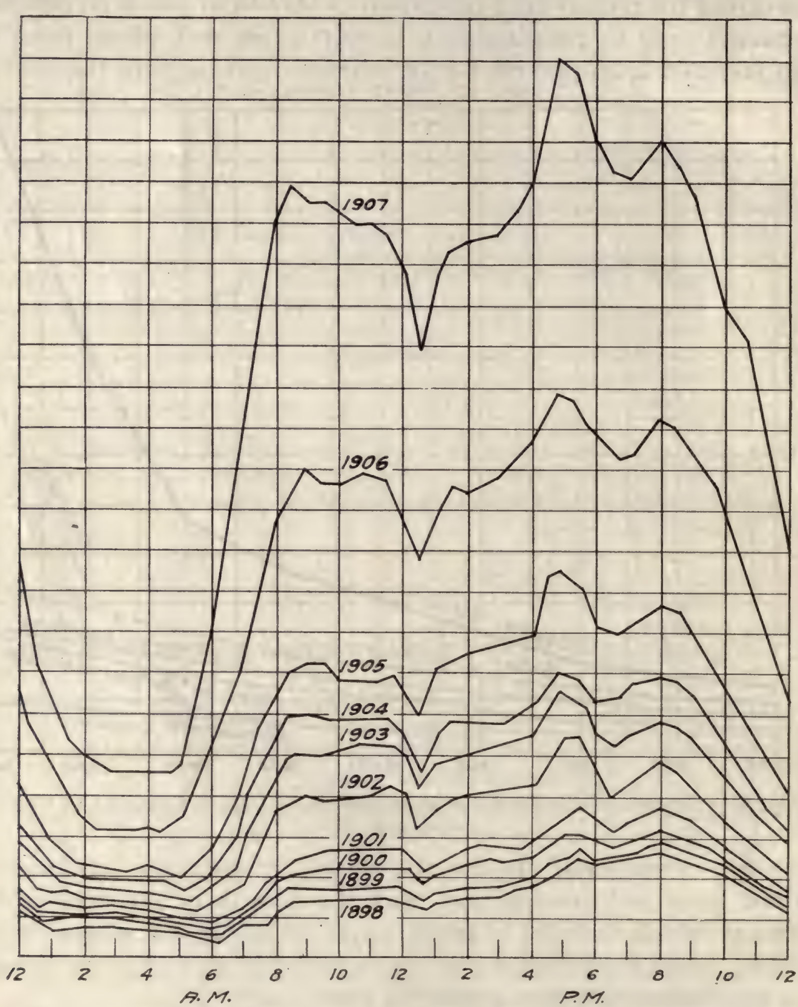


Fig. 31.—Average Daily Load Curve for Each Year of Commonwealth Edison Company of Chicago.

Electrical World, May 16, 1908). Fig. 31, shows the average load curve of the company for each year from 1898–99 to 1906–7 obtained by averaging the load at each hour in the day for the entire year from July to June inclusive. The variation between the

summer and the winter load curves as well as the growth in each is shown by Fig. 32 and Fig. 33 (from Western Electrician, May 16, 1908).

The growth in the power demand on the plants of the Hartford Electric Light Co. of Hartford, Connecticut, is shown in Fig. 34, page 58, which is a combined annual load curve for several years, and not only shows the increase in the electrical output of this system for

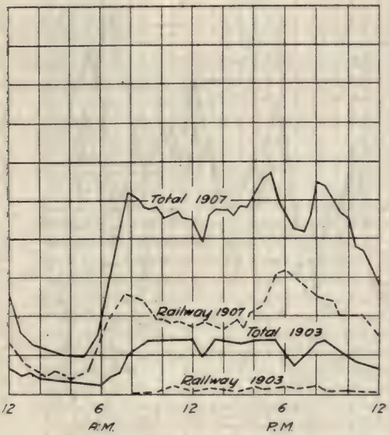


Fig. 32.—Summer Load Curves.

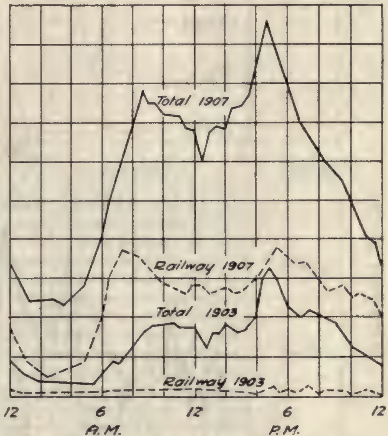


Fig. 33.—Winter Load Curves.

Curves Showing the Growth and Seasonal Daily Variations in Output of Commonwealth Edison Company of Chicago.

the years from 1898 to 1905, but also the annual monthly change in load from a maximum in December or January to a minimum in June or July. This variation fortunately accompanied similar variation in the flow of the Farmington river on which most of the power was developed.

Up to the middle of 1898 the entire load of this company was carried by a single water power plant. The natural increase in demand for power necessitated the construction of a second plant on the same river, and up to January, 1905, the two water power plants were able to carry most of the load, steam auxiliaries, however, being occasionally used, as indicated by the dotted line.

Figure 35, page 59, is a load curve of The London Hydraulic Supply Co., which is rather exceptional in that the power is used almost entirely for running elevators and is therefore almost exclusively a day load. The London Hydraulic Supply Company furnishes water under a pressure of 750 pounds per square inch through a system of

mains eighty-six miles long. In 1894, 2915 machines were connected to this system, of which 650 were passenger elevators, 2000 freight elevators and cranes, ninety presses of various kinds, ninety-five motors, and eighty fire hydrants. Each 1000 gallons of water pumped represents 8.738 H. P. hours, therefore, the maximum on the diagram represents about 1200 H. P. The preponderant influ-

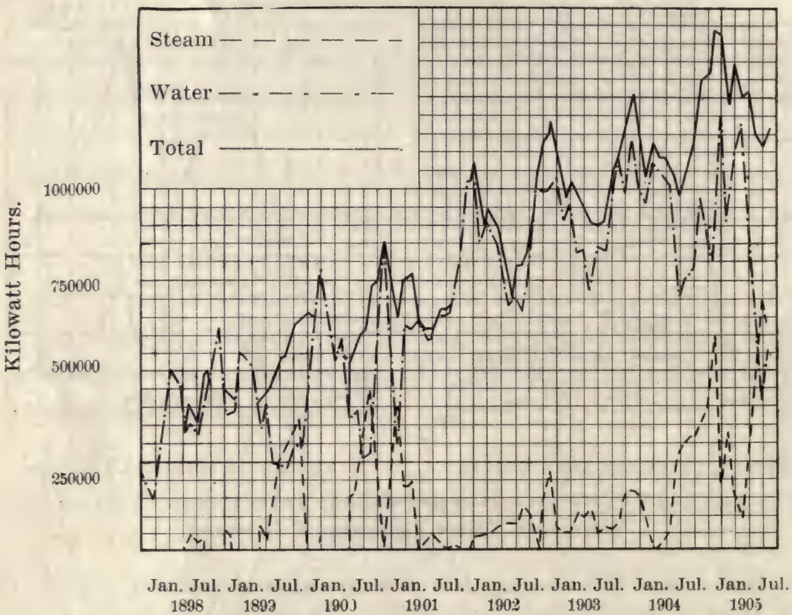


Fig. 34.—Energy Output of Hartford Electric Light Co. (from Electrical World and Engineer). (See page 57.)

ence of the elevator load is shown in the rapid rise from 6 to 10 A. M. and the somewhat slower decline from 4 to 12 P. M.

The growth of the service in seventeen years is indicated by the change in the load curve.

30. Load Conditions for Maximum Economy.—It is manifest that a plant will produce its maximum output if it be operated at full load for as much of the time as practicable. If it operates at less than full load, its output will be reduced and its income will also be reduced unless more is charged for the power delivered under such conditions. If the load carried for a large portion of the time is

comparatively small and the returns for such power are not proportionately large, the plant may for this reason be found unprofitable. On every plant the fixed charges, which include interest on first cost, depreciation charges and taxes, continue at a uniform rate every hour of the day and every day of the year. The operating expenses may increase somewhat with the total amount of power

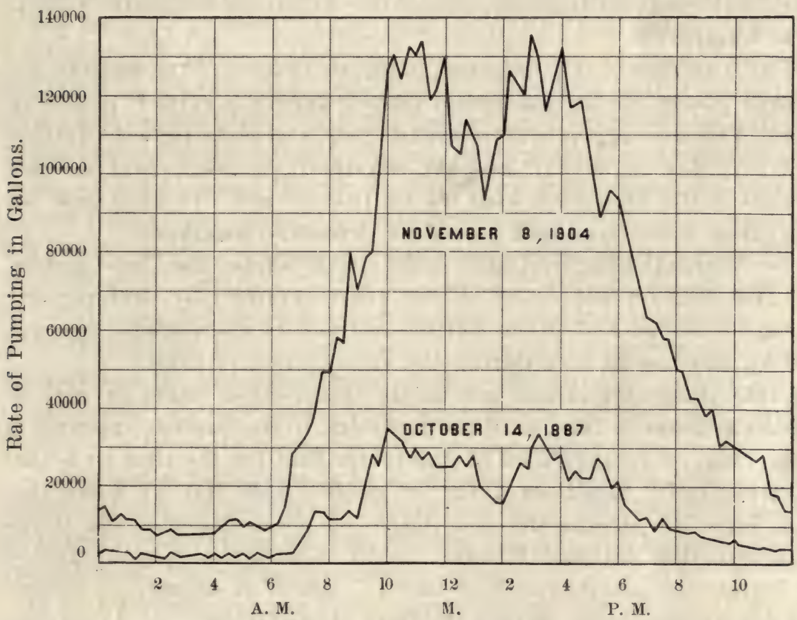


Fig. 35.—Maximum Days of Pumping—London Hydraulic Supply (Cassier's Magazine). (See page 57.)

furnished, but not in proportion. An increase in the total output of a given plant, if the unit charge for power be constant, means, therefore, a direct increase in the net earnings of the plant, and unless the power plant is constantly operating at its maximum capacity, its earning capacity is not at the highest point.

It will be noted at once that if a machine could be operated at its full capacity for the entire time, that the work done would be accomplished under the most economical conditions so far as each unit of output (horse power hour or kilowatt hour) is concerned. The interest on the first cost and other fixed charges will be chargeable to the maximum number of power units. The cost of wear

and of repairs, while increasing with the amount of power furnished, is not in direct proportion thereto and decreases per unit as the average load carried reaches nearer the maximum capacity of the machinery used. The same is true of the cost of attendance and most other operating expenses.

31. The Load Factor.—The term "load factor" is defined as follows, in the standardization rules of the American Institute of Electrical Engineers:

"The load factor of a machine, plant, or system is the ratio of the average power to the maximum power during a certain period of time. The average power is taken over a certain period of time, such as a day or a year, and the maximum is taken over a short interval of the maximum load within that period. In each case the interval of maximum load should be definitely specified."

The National Electric Light Association defines the "load factor" as "The fraction expressed in per cent. obtained by dividing the average load for any given period of time by the highest average load for any one minute during the same period of time."

While these definitions are easily understood, there is considerable variation in the use of the term due to its careless application to the ratio of average load to maximum load for the time of actual operation only, based on a ten or twelve hour day or a six day week, etc. To obviate the uncertainty of the meaning of the terms employed, the National Electric Light Association has suggested the term "operating-time load-factor," which it defines as a load factor considered only during the time of operation.

The use of these terms by engineers should be confined to their meaning as defined, or when used differently, the use should be clearly defined. As the term is defined, it bears no necessary relation to the total machine capacity of the plant nor to the capacity of the machines in actual operation, an expression for which sometimes seems desirable.

The ratio between the average load and the capacity of the machines used, may be termed the "machine factor."

When the plant has reached the growth for which it was designed, the "machine factor" and "load factor" on the day of maximum load would be the same if the load factor could be accurately predetermined and the machinery accurately selected for the same, there being of course a certain additional reserve capacity in the plant to take care of repairs, possible breakage and perhaps extra-

ordinary conditions. The ratio of the average load carried to the total capacity of the station is sometimes called the "capacity factor," which always must be less than the "machine factor," and "load factor," the amount depending upon the contingencies of operation and the necessity for a greater or less factor of safety.

The capacity of the machinery which must be installed in any power plant depends more largely on the maximum load that the plant must carry than on the total output of the plant. It is evident that a machine will produce the maximum output if it can be operated at its full load capacity to as great an extent as practicable; and on the other hand, the maximum output will be obtained under these conditions with the smallest investment for machinery.

32. **Effect of Load Factor on Necessary Plant Capacity.**—Water power plants are commonly designed to produce an approximately fixed output which depends in its amount on both the hydraulic resources and the market. This fixed amount may be an ultimate end to be attained rather than an immediate installation, and the immediate installation may be modified in amount both by the immediate market and the uncertainties as to how the ultimate output will be distributed. In any event, the distribution of load will greatly modify the ultimate necessities for machine capacity.

If a water power plant is to be installed to develop a total given output of say 90,000 H. P. hours per day, the selection of the machinery will depend on the relation between the highest and the average load it must be expected to carry.

If its load were equally distributed through every hour of the day and year, a minimum installation of machinery would be required, for the plant would be required to develop 3750 H. P. continuously, and would need a capacity only sufficient to carry this load and provide for the necessary reserve to take care of accidents and repairs. With such a continuous load, which is seldom if ever obtained, the load factor would be 100 per cent. If the same output must be delivered in half the time, or if the average load during twenty-four hours is 3750 H. P., and the peak load reaches 7500 H. P. for even a few minutes, twice the machinery for active operation, or 7500 H. P., would be required (see Fig. 36, page 62), and the load factor would be fifty per cent.

Table 6, page 63, shows the relations of the total active machine capacity necessary to develop the total output of 90,000 H. P. per day, with various load factors.

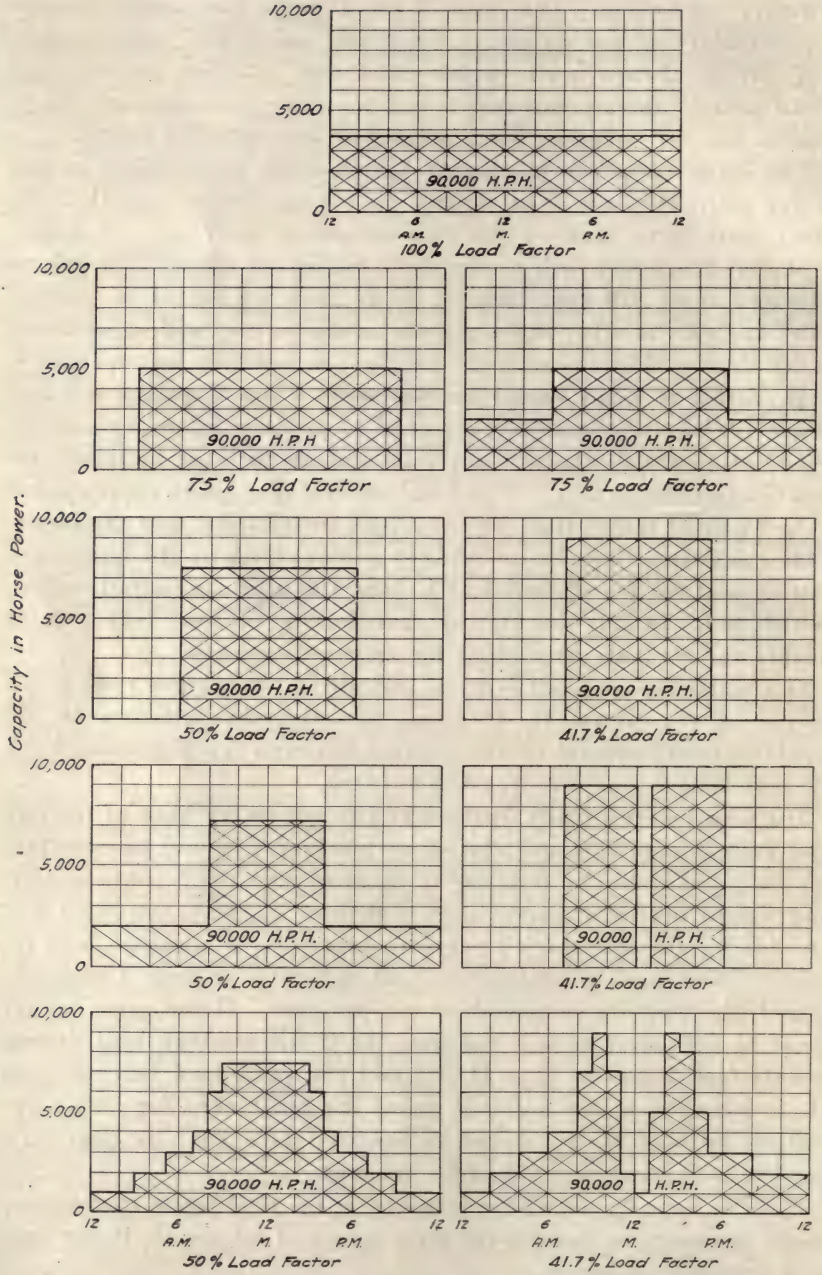


Fig. 36.—Relation of Machine Capacity to Load Factor (see page 61).

The second column shows the capacity of the plant required during the period of maximum load. To this a certain reserve capacity must commonly be added to provide for the necessary maintenance and repair of the machine.

The third column shows the number of hours that the machinery would operate at its maximum load to furnish the daily output of the plant.

TABLE 6.

Load Factor Per Cent	Active Horse Power Capacity of Machinery	Relative Hours of Use at Maximum Capacity
100	3750	24
75	5000	18
50	7500	12
41⅔	9000	10
33⅓	11250	8
25	15000	6

From Table 6 it is evident that the manner in which a given output is to be distributed will have a great effect on the design of a plant, and on the necessary cost of providing for such an output.

As the load factor is the ratio between average and maximum load, a given load factor may result from a great variety of load distribution (see Fig. 36, page 62). For the purpose of plant design, the approximate distribution of the load throughout the day, week, month and year must be known, estimated or assumed.

33. Effect of Load Factor and Load Curve on Machine Selection.— In order to have a plant work to the maximum advantage, it must be designed to fit the variations of the load so far as practicable. The operation of a machine at low loads is not only expensive on account of fixed charges, but is still more so on account of the decreased efficiency under such conditions (see Fig. 19, page 30).

In small plants with varying loads efficient operation usually involves the installation of two or more generators of such capacities that a single unit will efficiently furnish the power required during the hours of minimum demand. As the daily demand for power increases, additional units are started and operated, still under economical conditions, and at the peak of the load one or more additional units may be cut in and operated for the limited time during which the maximum demands prevail. Such an arrangement, properly carried out, assures efficiency of operation at all times, even when considerable changes of load are of hourly occurrence.

34. Influence of Management on Load Curve.—The relations of the "load curve," the "load factor," the "machine factor" and the "capacity factor" are, or may be, to an extent controlled by the business management of any plant, and by the selection and the character of the load to be carried, where such selection is possible. Each consumer of power will develop a particular load due to the character of the work done, and it is frequently possible, by a judicious selection of customers, and especially by a proper grading of rates, to raise the load factor and thereby decrease the cost of operation and increase the net profits from the plant. A study of the probable plant factors is necessary for the judicious selection of machinery in order to attain the most efficient operation and, in a hydraulic plant, in order to properly design it and conserve the maximum energy of the stream that is being developed.

35. Study of Load Curve in Relation to Stream Flow and Auxiliary Power.—Some of the relations between the load factor and the conditions under which a hydraulic plant may have to be operated are shown by Figs. 37, 38 and 39.

In Fig. 37, page 65, diagram A shows a typical daily load curve from the terminal station at St. Louis, a curve quite similar in general character to those previously shown.

Diagram B shows the power that must be developed from a stream in order to take care of the load represented by this load curve, under conditions where no auxiliary power or storage are available. In this case, it will be noted that the available water power must be equivalent to or greater than the maximum peak load, and that all power represented by the area above the load line, amounting in the case illustrated to about forty per cent. of the total available power, will be wasted.

Diagram C illustrates the condition where the average load and water power are equal. In this case, pondage or storage, represented by the cross-hatched area below the average line, may be utilized to furnish the peak power represented by the cross-hatched area above the average line. Without pondage, the cross-hatched area below the average load line will represent the energy of the stream wasted, and the cross-hatched area above the average load line will represent the energy which must be supplied by auxiliary power. Without pondage the power of the stream must be utilized as it passes, and in the diagram B, of Fig. 37, the power represented above the load line under such conditions must be wasted.

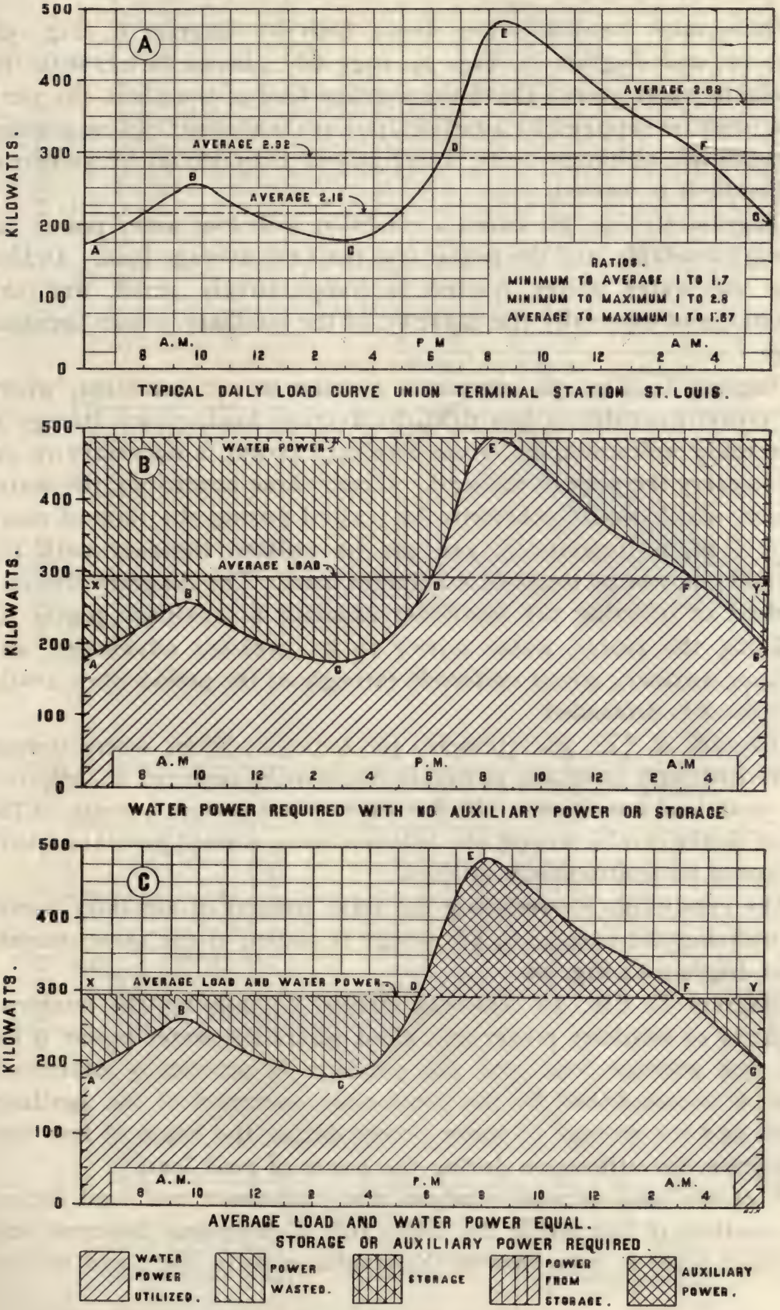


Fig. 37.—Relation of Power Supply and Demand (see page 64).

These same conditions are shown both by diagram C, Fig. 38, page 67, and diagram A, Fig. 39, page 68. In the latter, with the available water power above the average load of the plant, the peak load must be supplied by auxiliary power on account of low storage possibilities, although more water power than would be sufficient to handle it is wasted.

Diagram B, Fig. 38, shows a condition with low water power, no storage available, and the power less than the average load. In this case the water power wasted is comparatively small, and the amount, and especially the capacity, of the auxiliary power becomes large.

Diagram C, Fig. 38, represents a water power condition, where the power available is less than the average load, where storage is practically unlimited, and some auxiliary power is necessary in order to carry the peak of the load. Under these conditions, the water power, which would otherwise be wasted during the time of minimum load, is impounded, and can be utilized together with the auxiliary power at times of maximum load. The diagram shows a method of utilizing the minimum capacity of auxiliary power by utilizing the stored water power to its greatest advantage, and utilizing auxiliary power uniformly throughout the period when auxiliary power is demanded.

Diagram A, Fig. 39, represents the same conditions where storage is limited, and auxiliary power is necessarily required to help out the peak load conditions. In this case only a certain amount of the spare water can be stored, the balance being wasted at times where it cannot be continuously utilized.

The conditions for reducing the total amount of auxiliary power by utilizing the storage to advantage is shown in the same manner as in diagram C, Fig. 38.

Diagram B, Fig. 39, shows a method of utilizing the minimum capacity of auxiliary power in a plant where the water power is below the average load and the pondage is practically unlimited. This is accomplished by the continuous operation of the auxiliary plant and the storage of water power during the hours of low consumption, for utilization during the hours of peak load.

A careful and detailed study of the load curve and load factor; the method of increasing the latter and of designing the most economical plant to take care of the condition to be met; and the ad-

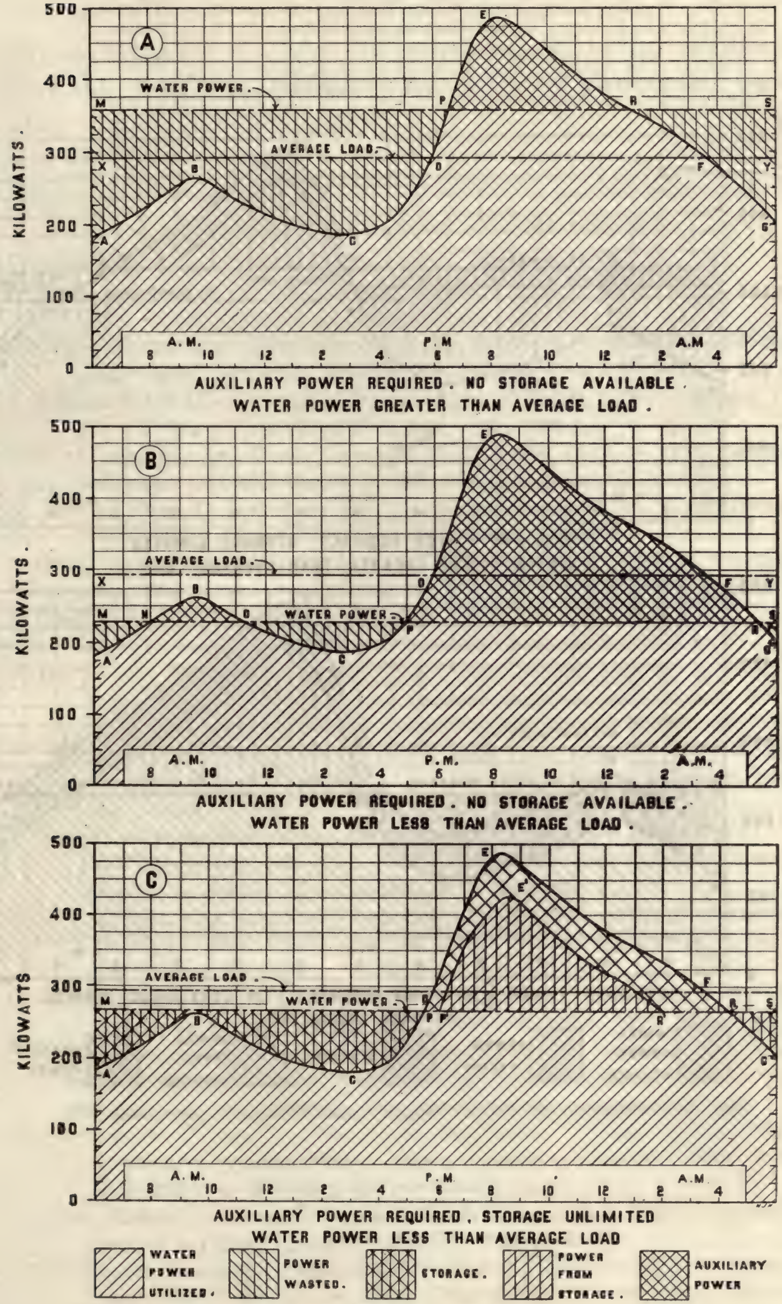


Fig. 38.—Relation of Power Supply and Demand (see page 66).

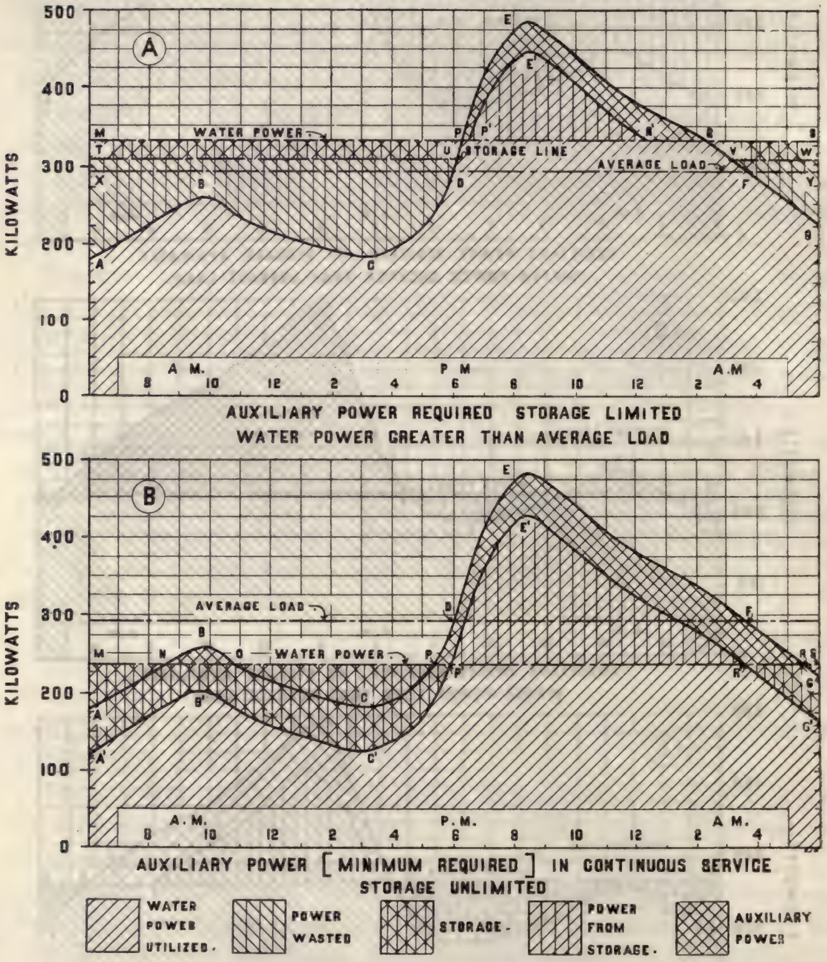


Fig. 39.—Relation of Power Supply and Demand (see page 66).

justment of rates to attain equitable returns to the investor at reasonable price to the consumer, are matters of plant design worthy of the best efforts of the engineer.

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CHAPTER IV

THE FLOW OF STREAMS

36. Symbols Used in This Chapter.—The letters and symbols used in this chapter have the following significance:

g = Acceleration due to gravity (32.2 feet per second per second).
l = Length of channel considered (lineal feet).
a, a' = Cross sectional area of water in the channel under normal and modified conditions respectively (square feet).
p, p' = Wetted perimeter of channel cross section under normal and modified conditions respectively (lineal feet).
r, r' = Hydraulic radius of the channel under the respective conditions noted $= \frac{a}{p}$ and $\frac{a'}{p'}$ respectively.
v, v' = Mean velocity of flow in the channel under the respective conditions noted.
q = Quantity of discharge in cubic feet per second $= av = a'v'$.
h_3, h'_3 = Friction head i. e. the fall in feet necessary to overcome the frictional resistance in the channel of length l while maintaining the velocities v and v' respectively.
s = Slope of channel = sine of the angle which the line of the hydraulic gradient makes with the horizontal $= \frac{h_3}{l}$
c, c' = Coefficient of friction for use in Chezy's formula (Equation 10) under normal and modified conditions respectively.
n = Coefficient of roughness for use in the formula of Ganguillet and Kutter.
m = Coefficient of roughness for use in Bazin's formula.

37. Laws of Uniform Flow in Channels and Conduits.—The flow in any channel may be considered on the basis of either of two principles.

First Principle: If a channel, pipe, or conduit of any length were free from friction, the flow in the same could be expressed by the formula

$$h_3 = \frac{v^2}{2g} \text{ or } v = \sqrt{2gh_3} \tag{6}$$

Friction is, however, always present and a friction coefficient must be introduced into this formula in order that it shall represent actual conditions.

Equation (6) thus corrected becomes

$$h_3 = c' \frac{v^2}{2g} \text{ or } v = c\sqrt{2gh_3} \tag{7}$$

Second Principle: In any channel, conduit, pipe or passage we may fairly assume:

First: That from axiomatic considerations *the resistance to the flow of water is directly proportional to the area of the surface in contact with the water.*

Second: That from observed conditions, *the resistance is found to be approximately proportional to the square of the velocity of flow.*

Third: That from experience, *the resistance to flow is inversely proportional to the cross-section of the stream.*

These conclusions may be expressed by the equation:

$$\text{Resistance} \propto \frac{(\text{Velocity})^2 \times \text{area of contact}}{\text{area of section}}$$

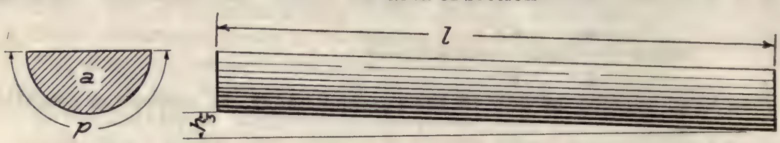

Fig. 40.—Relations of Area, Wetted Perimeter and Slope in a Uniform Channel.

The area of the surface of a channel is the product of the wetted section or wetted perimeter p times the length of the section, or, $p \times l$ (see Fig. 40). The velocity is represented by v and the cross-section by a. Hence, from the above considerations, we may write for the friction head:

$$(8) \qquad h_3 \propto \frac{v^2 pl}{a} \text{ and by transposition } v^2 \propto \frac{ah_3}{pl}$$

That is to say, the square of the velocity is in direct proportion to the area of the section and to the friction head and inversely proportional to the wetted perimeter and to the length of the section.

In practice it is found that there are numerous factors which affect the theoretical conditions, as above set forth, which must therefore be modified in accordance with the conditions which obtain. In formula (8) therefore it is necessary to apply a coefficient (c') which represents the summation of such other influences. The form in which this last equation is ordinarily written is

$$(9) \qquad h_3 = c' \frac{v^2 pl}{a} \text{ or } v = c \sqrt{\frac{ah_3}{pl}}$$

Ordinarily this form is somewhat abbreviated by substituting for a/p the hydraulic radius which represents this ratio. That is to say,

$$\frac{\text{area of cross-section}}{\text{wetted perimeter}} = \frac{a}{p} = r$$

The "hydraulic radius" is also sometimes termed the "mean depth" or the "mean radius." For the ratio of the resistance head to the length of section the equivalent slope s is substituted.

That is to say:

$$\frac{\text{Resistance head}}{\text{Length of section}} = \frac{h_3}{l} = s$$

With these substitutions the formula (9) assumes the final form of:

$$(10) \qquad v = c\sqrt{rs}$$

From this it follows that as the quantity of water flowing in the channel will be equal to the area multiplied by the mean velocity

$$(11) \qquad q = av = ac\sqrt{\frac{ah_3}{pl}} = ac\sqrt{rs}$$

In the use of this formula three factors must be determined by measurement or estimate in order to derive the fourth. v, r and s can be determined experimentally or measured directly. The factor c is the most difficult to ascertain as it depends upon a very great variety of conditions which can only be appreciated and estimated by means of a knowledge of the conditions under consideration, and by comparison of such conditions with similar observed conditions. The degree of accuracy with which c can be estimated depends largely upon intelligent experience, that is upon a knowledge of what that value has actually been found to be under conditions similar to those under consideration and on which an estimated value is required. Various attempts have been made to derive a formula which will give the approximate value of c in accordance with the varying conditions and independently of individual experience.

The principal formulas for the values of c are those of Ganguillet and Kutter and of Bazin. Ganguillet and Kutter's formula for the value of c is as follows:

38. Kutter's Formula.—

$$(12) \qquad c = \frac{41.6 + \frac{1.811}{n} + \frac{0.00281}{s}}{1 + \left(41.6 + \frac{0.00281}{s}\right)\frac{n}{\sqrt{r}}}$$

From this formula it will be seen that Ganguillet and Kutter assume c to vary with the slope, with the square root of the hydraulic

radius and with a new factor *"n"* which is termed the coefficient of roughness. The value of this coefficient as determined by these experiments is as follows:

For large pipe with the following characteristics:

Exceptionally smooth cast iron pipe	n = .011
Ordinary new cast iron or wooden pipe	.0125
New riveted pipes and pipes in use	.014
Pipes in long use	.019

For open channels of uniform sections:

For planed timber sides and bottom	n = .009
For neat cement or glazed pipe	.01
For unplaned timber	.012
For brick work	.013
For rubble masonry	.017
For irregular channels of fine gravel	.02
For canals and rivers of good section	.025
For canals and rivers with stones and weeds	.030
For canals and rivers in bad order	.035

39. **Bazin's Formula.**—It has been questioned by many observers whether the slope of the channel has any material influence on the value of the coefficient *c*. Bazin has derived a formula based on his examination of this subject in which he assumes that *c* does not vary with the slope. His formula, which is intended for the calculation of flow in open channels is shown, together with a graphical table based thereon, in Fig. 41, page 75. This figure illustrates the law of variation of *c* and is applicable in principle in a general way to all channels and passages.

40. **Efficiency of Section.**—It will be noted from the previous equation

$$q = va = ca\sqrt{rs} = ca\sqrt{\frac{ah_3}{pl}} \tag{11}$$

that with c and s constant q varies as $a\sqrt{r}$ or as $\sqrt{\frac{a^3}{p}}$

From this the conclusion may be drawn that other things being equal the maximum quantity of water will pass through any section of any river or other channel in which the hydraulic radius is a maximum or the wetted perimeter a minimum. Where a choice exists as to the class of material with which the channel is to be lined *c* becomes a variable and *q* will vary as

$$ca\sqrt{r} \text{ or as } c\sqrt{\frac{a^3}{p}}$$

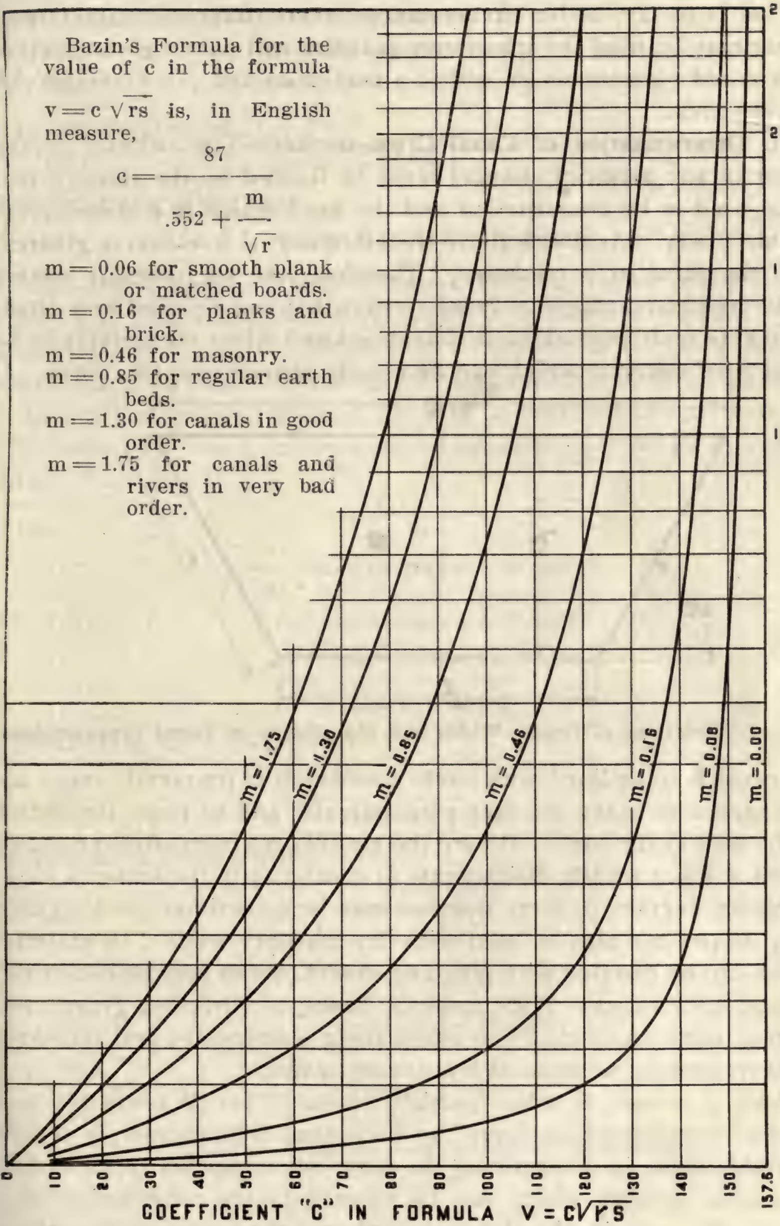

Fig. 41.—Diagram For Solution of Bazin's Formula (see page 74).

That is to say, under circumstances where different characters of lining may be used the maximum quantity will pass a given section with c and r maximum or with c a maximum and p a minimum for a given area.

41. Determination of Canal Cross-section.—The velocity of the water in any artificial channel must be limited by the class of material used in its construction and the head which it is found practicable to use. As noted above the efficiency of a section is greatest with the value of p minimum. Therefore, the semi-circular section is the most advantageous cross-section that can be used in a channel where resistance alone is considered and when the canal is to be lined with material which can be readily shaped into this form. If

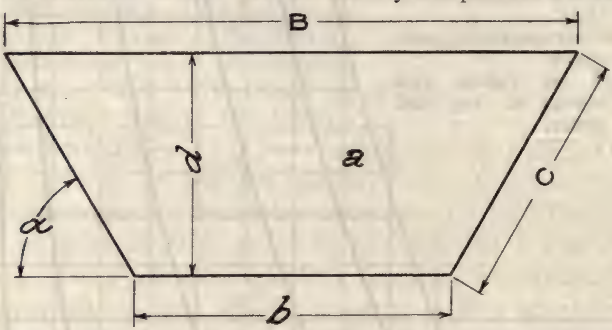

Fig. 42.—Relations of Depth, Width and Side Slopes of Canal Cross-sections.

the canal is to be lined with stone masonry it is frequently more advantageous to make the face perpendicular and to place the batter of the wall at the back. Where the canal is cut from stone or shales which will not readily disintegrate in contact with the water, a slope of ninety degrees to forty degrees may be sometimes used. Quite steep slopes can also be used with dry masonry walls. In material which can be handled with pick and shovel, slopes may be used from 1 : 1.25 to 1 : 1.50. With artificial banks of dirt and gravel and in most earth channels a less slope angle is necessary and the slope must frequently be made as low as one to two.

Table 7, which is taken partially from "Uber Wasserkraft und Wasser Versorgungsanlagen," by Ferdinand Schlotthauer, is of considerable value in determining the most advantageous cross-section in various sections which may be adopted in the construction of a canal. As seen in the discussion above, the most advantageous cross-section, other things being equal, is that in which the wetted perimeter is a minimum or the hydraulic radius is a maximum. The following general discussion of the relations is based on Fig. 42.

From this figure it will be seen that

$$(13) \qquad a = bd + d^2 \cot \alpha$$

$$(14) \qquad p = b + 2d \operatorname{cosec} \alpha$$

The transposition of (14) gives

$$(15) \qquad b = p - 2d \operatorname{cosec} \alpha$$

Substituting (15) in (13)

$$(16) \qquad a = dp - 2d^2 \operatorname{cosec} \alpha + d^2 \cot \alpha$$

The above equation now contains the area, depth, wetted perimeter and functions of the slope angle, in this case a constant. The conditions of maximum efficiency of a canal section require that the wetted perimeter be a minimum or what amounts to the same thing with a given wetted perimeter the area *a* must become a maximum. The value of *d* which makes *a* the maximum is determined by putting

$$\frac{d(a)}{d(d)} = 0$$

$$(17) \qquad \frac{d(a)}{d(d)} = p - 4d \operatorname{cosec} \alpha + 2d \cot \alpha$$

$$(18) \qquad 0 = p - 4d \operatorname{cosec} \alpha + 2d \cot \alpha$$

$$(19) \qquad d = \frac{p}{4 \operatorname{cosec} \alpha - 2 \cot \alpha}$$

Substituting for p its value in (14)

$$(20) \qquad d = \frac{b + 2d \operatorname{cosec} \alpha}{4 \operatorname{cosec} \alpha - 2 \cot \alpha}$$

Equation (13) transposed reads

$$(21) \qquad b = \frac{2 - d^2 \cot \alpha}{d}$$

Substituting this value in (20) we have

$$(22) \qquad d = \frac{\frac{a}{d} - d \cot \alpha + 2d \operatorname{cosec} \alpha}{4 \operatorname{cosec} \alpha - 2 \cot \alpha}$$

Clearing:

$$(23) \qquad 4d^2 \operatorname{cosec} \alpha - 2d^2 \cot \alpha = a - d^2 \cot \alpha + 2d^2 \operatorname{cosec} \alpha$$

Transposing:

$$(24) \qquad d^2 = \frac{a}{2 \operatorname{cosec} \alpha - \cot \alpha}$$

Transforming trigonometric functions

$$(25)\qquad d^2 = \frac{a}{\dfrac{2}{\sin \alpha} - \cos \alpha \operatorname{cosec} \alpha}$$

$$(26)\qquad = \frac{a}{\dfrac{2 - \sin \alpha \cos \alpha \operatorname{cosec} \alpha}{\sin \alpha}}$$

$$(27)\qquad = \frac{a \sin \alpha}{2 - \cos \alpha}$$

Finally:

$$(28)\qquad d = \sqrt{\frac{a \sin \alpha}{2 - \cos \alpha}}$$

Equation (21) may be written

$$(29)\qquad b = \frac{a}{d} - d \cot \alpha$$

Table 7 is calculated from the formulas:

$$(28)\qquad d = \sqrt{\frac{a \sin \alpha}{2 - \cos \alpha}}$$

$$(30)\qquad b = \frac{a}{d} - d \cot \alpha$$

$$(31)\qquad B = b + 2d \cot \alpha$$

$$(32)\qquad p = b + \frac{2d}{\sin \alpha}$$

In the above, a = cross-section area; d = depth of water in channel; b = bottom width; B = width at water level; p = wetted perimeter; c = the length of slope which is equal to $\dfrac{d}{\sin \alpha}$

In Table 7, page 79, the relation of these functions for the slopes ordinarily used in practice have been calculated as well as for the semi-circular section. The use of the table may be illustrated as follows: The quantity of water which it is desired to deliver is determined by the conditions of the problem or by measurement. The velocity to be maintained in the channel is determined by the existing slope, the nature of material encountered, or the friction head which it is found desirable to maintain. The area of

TABLE 7.

Economic Dimensions of Canal Cross-Sections with Various Slopes. Dimensions Expressed as Functions of Area of Cross-Section (see page 76).

Side Slope.	Slope Angle.	Relative Slope. cot α	Depth. d	Bottom Width. b	Width of Water Surface. B	Length of Slope Line. C	Wetted Perimeter. p	Hydraulic Radius. r	Character of Material to Which Side Slope is Adapted.
Vertical ..	90°00′	0.0000	$.707\sqrt{a}$	$1.414\sqrt{a}$	$1.414\sqrt{a}$	$.707\sqrt{a}$	$2.828\sqrt{a}$	$.353\sqrt{a}$	Concrete and Stone Masonry.
1:1¾	60°15′	0.5714	$.760\sqrt{a}$	$.882\sqrt{a}$	$1.750\sqrt{a}$	$.875\sqrt{a}$	$2.633\sqrt{a}$	$.380\sqrt{a}$	Dry Masonry Walls.
1:1	45°00′	1.0000	$.740\sqrt{a}$	$.613\sqrt{a}$	$2.093\sqrt{a}$	$1.046\sqrt{a}$	$2.705\sqrt{a}$	$.369\sqrt{a}$	Clayey Gravel or Hardpan.
1¼:1	38°40′	1.2500	$.716\sqrt{a}$	$.503\sqrt{a}$	$2.293\sqrt{a}$	$1.146\sqrt{a}$	$2.795\sqrt{a}$	$.358\sqrt{a}$	Well Compacted Clay.
1½:1	33°41′	1.5000	$.689\sqrt{a}$	$.418\sqrt{a}$	$2.485\sqrt{a}$	$1.243\sqrt{a}$	$2.904\sqrt{a}$	$.344\sqrt{a}$	Coarse Gravel, Clayey Loam.
2:1	26°34′	2.0000	$.636\sqrt{a}$	$.300\sqrt{a}$	$2.844\sqrt{a}$	$1.422\sqrt{a}$	$3.144\sqrt{a}$	$.318\sqrt{a}$	Ordinary Earth, Coarse Sand.
2½:1	21°48′	2.5000	$.588\sqrt{a}$	$.230\sqrt{a}$	$3.170\sqrt{a}$	$1.584\sqrt{a}$	$3.398\sqrt{a}$	$.294\sqrt{a}$	Loose Earth.
3:1	18°26′	3.0000	$.549\sqrt{a}$	$.173\sqrt{a}$	$3.471\sqrt{a}$	$1.738\sqrt{a}$	$3.649\sqrt{a}$	$.274\sqrt{a}$	Loose Earth, Sand.
Semi-Circular	....	....	$.798\sqrt{a}$	....	$1.596\sqrt{a}$	....	$1.773\sqrt{a}$	$.563\sqrt{a}$	Brick, Concrete, Metal.

the cross-section required to carry the quantity q with velocity v is $a = \frac{q}{v}$. After the slope angle has been selected, for the material in which the channel is to be constructed, the corresponding values may be taken out of the table from their respective columns and multiplied by the square root of a. The result thus obtained gives the desired dimensions. If, for example, we desire to carry 100 cubic feet of water per second in a canal at a velocity of two and one-half feet per second at which velocity small pebbles are unaffected, and a side slope of one and five-tenths to one, which is suitable for earth, has been decided upon, the required area of cross-section will be $100/2.5 = 40$ square feet. The square root of 40 is 6.33. The required dimensions of canal as taken from the table are

Depth $d = .689 \times 6.33 = 4.36$ ft.
Bottom width $b = .418 \times 6.33 = 2.65$ ft.
Top width $B = 2.485 \times 6.33 = 15.73$ ft. and
The wetted perimeter $p = 2.904 \times 6.33 = 18.38$ ft.

Computation of the area from the above dimensions gives forty square feet. Hence the work has been checked.

In the actual design of canals, other factors beside the most efficient cross-section will have an important bearing. The presence of rock or ground water in the section, the necessity of securing sufficient suitable material for embankment and various other conditions, may make some other than the most efficient section desirable in any particular case. *The efficient section is therefore an ideal to be approximated and must be modified by other practical considerations.*

42. **Flow in River Channels.**—The previous discussion of the flow of water in channels applies only to such channels as have uniform cross-sections, alignment, and gradient and a bed of uniform character throughout the length considered. Such conditions may be closely approximated in artificial channels in which the quantity of water flowing is under control. In such channels, and with a steady flow, that is with the same quantity of water passing every cross-section in the same time, equations (10) and (11) are found to fairly represent the conditions of flow.

In natural water courses no two cross-sections are the same and the area a, and wetted perimeter p, and the fall h, in any length l, usually differ considerably from reach to reach. The quantity q, of water flowing in any such stream is also constantly changing. There every condition of uniform flow is lacking and can only be

approximated for selected reaches of such streams and during periods when stream flow is fairly steady.

43. Changes in Value of Factors with Changes in Flow.—From an examination of equation (11) it is evident that in any channel as the quantity of water flowing, *q*, changes, there must be a corresponding change in some or all of the factors on the other side of the equation.

For steady flow in a uniform channel, *s* remains constant and all changes are confined to the values of *a, c* and *r*. The laws of change in the values of *c* are given by Kutter's and Bazin's formulas, but are best illustrated and understood by reference to Fig. 41, which is a graphic expression of the formula of Bazin.

In variable flow a change in all of the factors usually accompanies a change in the value of *q*, each factor changing in accordance with the physical conditions of the channel.

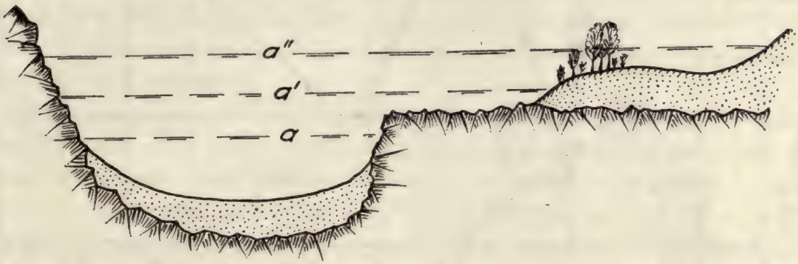

Fig. 43.—Variation in Hydraulic Conditions with Change in Stage of River.

The changes in the value of *c*, in an irregular channel, may not and usually do not follow closely the laws expressed by Kutter's or Bazin's formulas. This is due principally to the fact that an increase in the depth of water in a channel may be accompanied by a radical change in both the character of the friction area of contact added to the stream bed, and the shape of the cross-section (see Fig. 43).

In such cases *c* is even found to decrease as *r* increases. The law of simultaneous increase in *c* and *r* presupposes a channel of uniform character and condition. If an increase in the hydraulic radius *r*, in any channel is accompanied by a radical change in the character and shape of its bed the general law will not hold. It is evident that under such conditions where the character and bed are not relatively similar the values of *c* for different values of *r* may not be fairly comparative. No more uniform law of change can be

expected under such conditions than would occur in the comparison of the relation of *c* and *r* for entirely different channel sections.

In Fig. 44 are shown the observed values of *c* and *r* for certain reaches of the Wisconsin River above Kilbourn, Wisconsin. It will be noted

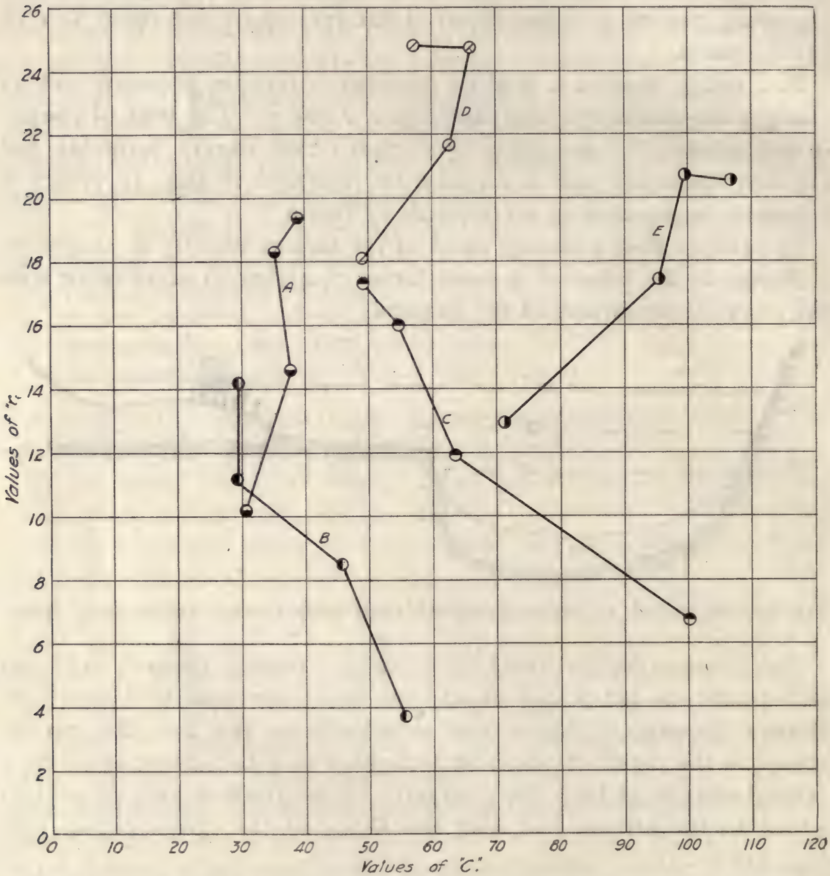

Fig. 44.—Relations of Coefficient to Hydraulic Radius in Certain Reaches of the Wisconsin River.

that the value of *c* for reaches *A, D* and *E* decrease with the decrease of *r* and roughly approximate the law established by Bazin. On the other hand the value of *c* and *r* for reaches *B* and *C* increase with a decrease in *r,* a condition due to irregularities in the cross-section of the reach.

Where the values of a, p and r vary radically from section to section and differ materially from the values in the sections considered and on which calculations are based, the value of c will be found to differ considerably from that which the character of the bed and the entire section would indicate. Absurd values of c are a clear indication that the sections selected are not representative. The calculated value of c is modified by all unknown or unconsidered factors of the reach. The influences of irregularities in bed or section, the presence of unconsidered bends or changes in the gradient, and all other irregularities in the channels, modify the values of c.

44. Effects of Variable Flow on the Hydraulic Gradient.—In order to understand the effect of variable flow on the surface gradient of a stream, and in order to realize how conclusions drawn from the laws of uniform flow must be modified to meet conditions found in natural streams, it is necessary to consider the cause of variable flow in a stream, the variation in channel conditions, and both the effect of flow on such conditions and the effect of such conditions on the flow of a stream.

The surface of a stream is constantly fluctuating, not only on account of the variation in flow, but also on account of wind, barometric pressure and changes in the hydraulic gradient. Such changes occur from hour to hour, and even from minute to minute. Larger rivers, fed directly by great lakes, are more susceptible to these changes on account of the broad lake area, giving wind and barometric pressure greater opportunity to act. Every stream is, however, more or less susceptible to these changes, and gauge readings taken daily, therefore, show only in an approximate way the true height of the surface of the river at the point of observation. This is well shown by Fig. 45, page 84, which is reproduced from the autographic record of a gauge at the head of the St. Clair River.

45. Effects of a Rising or a Falling Stream on Gradient.—In a channel of uniform section, the bed of the channel AB (see diagram A, Fig. 46, page 85) having a uniform slope, all cross-sections, such as Aa and Bb, will be alike and the wetted perimeters and the hydraulic radii will be identical for all sections. The fall, bx, will be uniform in all equal lengths l, of the channel, and such uniform conditions will be maintained for all regular discharges after regular flow is once established.

In such channels, during changes in the stages of flow, the hydraulic gradient or slope will change until uniform flow is estab-

lished. In all cases illustrated in Figs. 46, page 85, and 47, page 88, the line *ab* represents the hydraulic gradient which will obtain if uniform flow is maintained in the channel and if there be no change in the channel section or other conditions. The actual water surface, caused by variable flow, is in each case shown by the line *a′b*. In each case, the fall, *bx*, would be necessary to produce uniform flow from *A* to *B* and to assure the flow of the normal quantity of water passing the section *Bb* as in diagram A. In diagrams B and C, Fig. 46 the conditions of variable flow in a uniform channel are graphically represented. The actual flow is greater or less than the normal quantity, according as the gradient is increased or diminished.

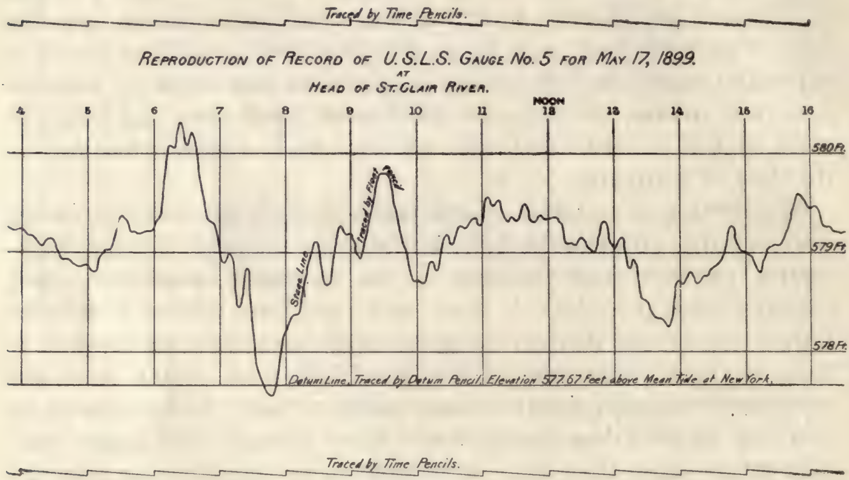

Fig. 45.—Variations in Gauge Height of the St. Clair River (see page 83).

In diagram B, the conditions with a rising stream are shown. Under these conditions the quantity of water passing the section *Aa* is greater than the quantity passing the section *Bb*, by the quantity of water necessary to fill up the channel of the stream to a new and uniform surface gradient. The head needed to produce the flow past the section *Aa*, is represented by the height *xx′*. The total fall between *A* and *B* is therefore greater than that required for the uniform flow as represented by the head *bx′*. This produces not only a greater flow at *Aa*, but also a flow greater than would be normal at section *Bb*.

In diagram C, Fig. 46, page 85, the conditions of a falling stream are represented. In this case, the depth at section *Bb* at the moment

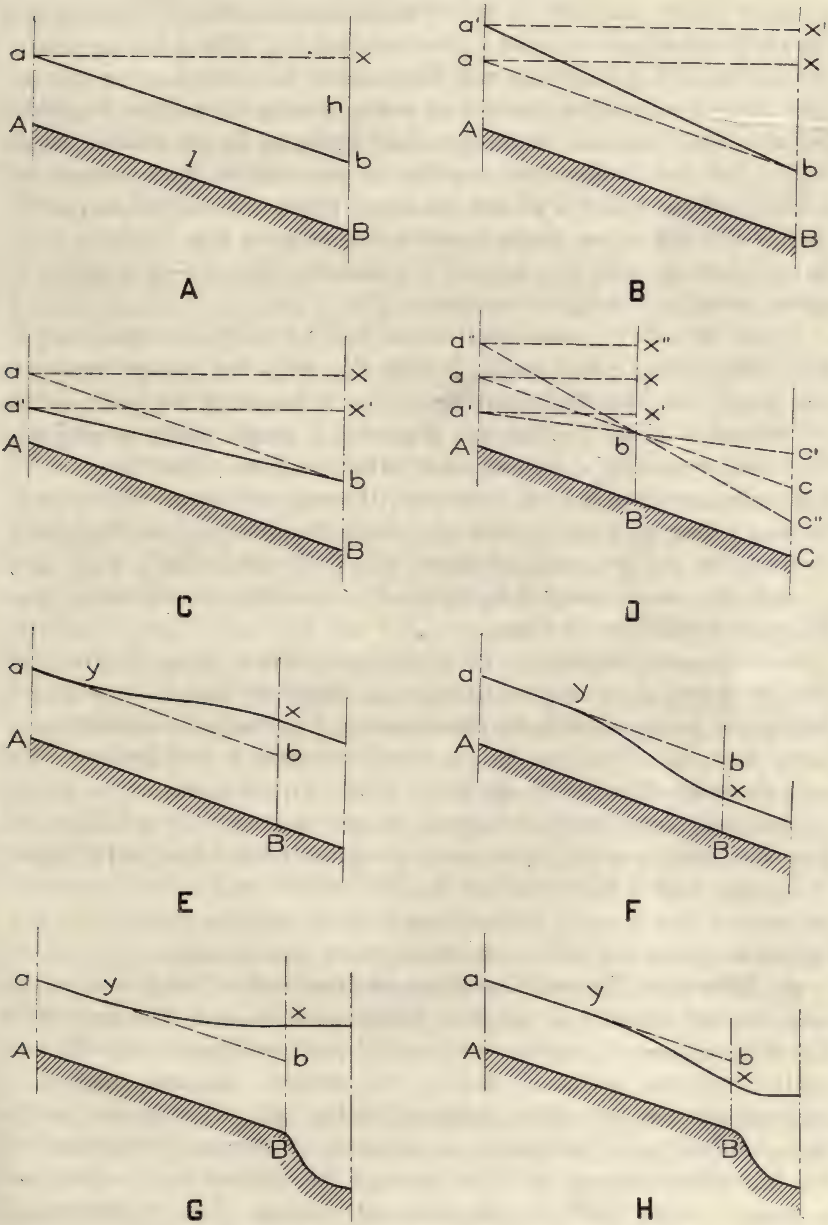

Fig. 46.—Effects of Variable Flow and Channel Conditions on the Hydraulic Gradient of a Stream (see page 83).

of observation would, if the flow was uniform, represent a normal flow which would require the fall bx, to maintain it. With a falling stream, the section AB is emptying and the quantity of water passing the section Aa is less than the quantity of water passing the section Bb, which in turn is also less than the normal flow indicated by the existing depth. A less fall bx', is therefore required to produce the flow passing Bb, which, with the lower slope and the same cross-section is less in quantity than would be the case under conditions of uniform flow. This fall bx', is less than the fall bx, required for uniform flow by the height xx': consequently the slope of the river is $a'b$.

From the above considerations it will be seen (see diagram D, Fig. 46) that a given gauge height Bb, may not always represent the same flow, for the discharge Q, is a function not only of the cross-section a, but also of the slope s. A single gauge height may therefore represent a considerable range of flows depending on the hydraulic gradient which may pass through the point with a uniform, a rising or a falling stream. It is obvious that the flows represented by the hydraulic gradient $a'bc'$, abc and $a''bc''$, while producing the same gauge height at Bb, nevertheless represent three different conditions of flow.

In the establishment of the relations between gauge heights and flow, it is therefore important that the observed flow corresponding to a given gauge reading be taken during a period of essentially uniform flow, for, from the above considerations, it will be seen that any determination or observation made with a rising or a falling stream must necessarily be more or less in error. It will also follow that, after a rating curve and rating table have been established, the gauge height taken during changes in the conditions of flow will be more or less in error, although such errors will compensate to a considerable extent and will, in the main, prove unimportant.

46. Effects of Channel Condition on Gradient.—The flow of water in a natural channel is far from being uniform and it is important for the engineer to realize this lack of uniformity and the effect of such conditions upon the flow of the stream. In any channel of uniform gradient, as AB in diagram E (Fig. 46), if at the section Bb the coefficient c is decreased on account of increased roughness in the bed of the stream, or if the area of the channel a, is contracted, a change in the hydraulic gradient will follow. The normal gradient with uniform flow would take the position ab, but on account of the change in conditions at Bb, the depth must increase to keep q

a constant; a must increase to offset the decrease in c or c must increase to offset the decrease in a if q remains constant. The surface must therefore rise to the point x and a new hydraulic gradient will be established and maintained until other changes in the channel condition again modify the same. Between the new and old gradients, a transition curve will be established extending both above and below the point at which the change in condition takes place to some point y, frequently a long distance upstream.

The opposite condition is shown by diagram F, Fig. 46, page 85. In this diagram the effect of an increase in the coefficient c, of the bed or in the area a, of the stream is represented. If c increases, a less section will be required below that point and again the surface is lowered; or if the width of the stream increases, the depth will diminish in order that ca may remain constant.

Variable flow is also caused by a sudden enlargement in the river section or by a discharge of the stream into a larger stream or into a lake or pond. Such conditions are shown by diagrams G and H, Fig. 46. The character of the transition curve in such cases will depend on the height of the surface of the water into which the stream is discharged. If the water surface of the lake is above b, the curve will be concave upward (see diagram G) and if the surface is below b, the curvature will be downward (see diagram H).

47. Effect of Change in Grade and of Obstructions.—Variable flow may also be caused by changes in the slope of the stream bed as shown by diagrams A and B, Fig. 47. The area of the stream must increase as the bed slope is decreased, or must decrease as the slope of the bed is increased in order to fulfill the conditions of equation (11).

It is evident that uniform slope may be maintained even with changed conditions if the changes that occur give rise to equal and opposite effects. For example, uniform slope may be maintained if the area of section a is reduced and the coefficient c is increased to such an extent that the product ac remains constant at each section of the channel.

Variable flow is also caused by the passage of the stream over weirs or dams and the effect on the gradient will vary as shown by diagrams C and D, Fig. 47, page 88. Variations may also be caused by a change in the bed (see diagram E, Fig. 47), or by local contractions, submerged weirs or other obstructions as shown by diagram F. Fig. 47.

In all of the above described cases it is obvious that if the slope of the stream is measured on any of these transition curves, a false idea of slope will obtain and a false relation will be established for

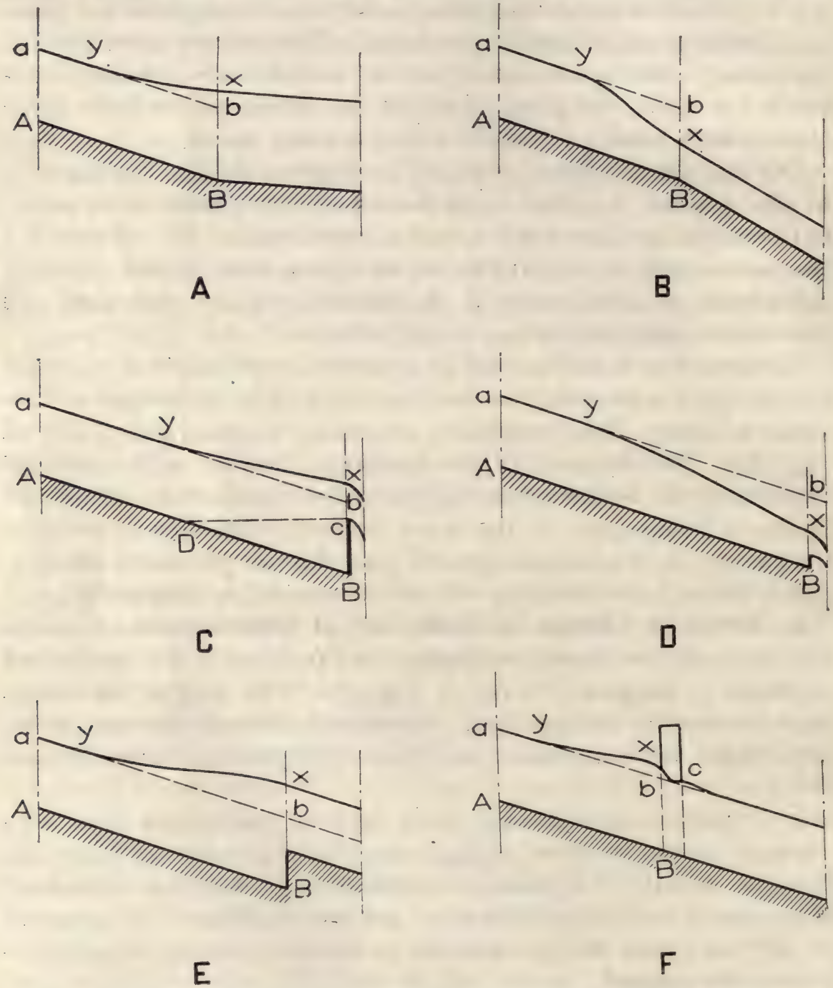

Fig. 47.—Effects of Channel Grade and of Obstruction on the Hydraulic Gradient of a Stream (see page 87).

the condition of stream flow. It is therefore essential in any measurement of a stream or in the establishment of any gauging station that the location for such observations be carefully selected on a reach of the stream where conditions of essentially uniform flow

prevail and that all observations be taken during stages where the flow of the stream is practically constant. If gauges are established at various points along the course of a river and are read simultaneously, and if the flow is uniform and no falls, rapids or tributaries intervene, the same differences in elevation should always obtain with the same stage of water.

A system of gauges as described above was established at Kilbourn on the Wisconsin River in order to determine the river slopes near that place. A large number of practically simultaneous readings were taken in order to determine the relations between the gauge heights at the various points compared with the Kilbourn gauge.

Figure 48, page 90, shows the results of the gauge readings at the various stations compared with the gauge readings at Kilbourn. It will be noted from the diagram that the slope of the river was far from uniform at different times during these readings, and, in a number of cases, the same gauge reading at Kilbourn was accompanied by readings at other gauges that differed from each other by more than a foot. For example, compare the gauge readings at Kilbourn with the readings at gauge No. 5. With a gauge reading of seventeen feet at Kilbourn, the normal gauge reading at No. 5 should be twenty-three feet, and with a normal flow, the fall between gauge No. 5 and the Kilbourn gauge would be five feet. From the diagram it will be seen that during a certain stage of flow in the river the gauge reading at gauge No. 5, with a seventeen foot reading at Kilbourn, was about twenty-two and one-half feet. Under these conditions the fall between gauge No. 5 and the Kilbourn gauge was only four and one-half feet. The slope being reduced, the quantity of water actually passing the Kilbourn gauge under these conditions was less than the normal flow for the seventeen feet gauge height. On two other occasions where the gauge reading at Kilbourn was approximately seventeen feet, the actual gauge reading at gauge No. 5 was about twenty-four feet. During these conditions the actual fall in the river between gauge No. 5 and the Kilbourn gauge was five feet, or one foot more than normal. Hence the quantity of water flowing by the Kilbourn gauge at this time was more than the normal quantity indicated by the Kilbourn gauge.

Readings of other gauges compared with the Kilbourn readings will show that at certain times the flow was normal and at other times the river must have been rising or falling and that consequently the gauge at Kilbourn at the time of such reading, was not

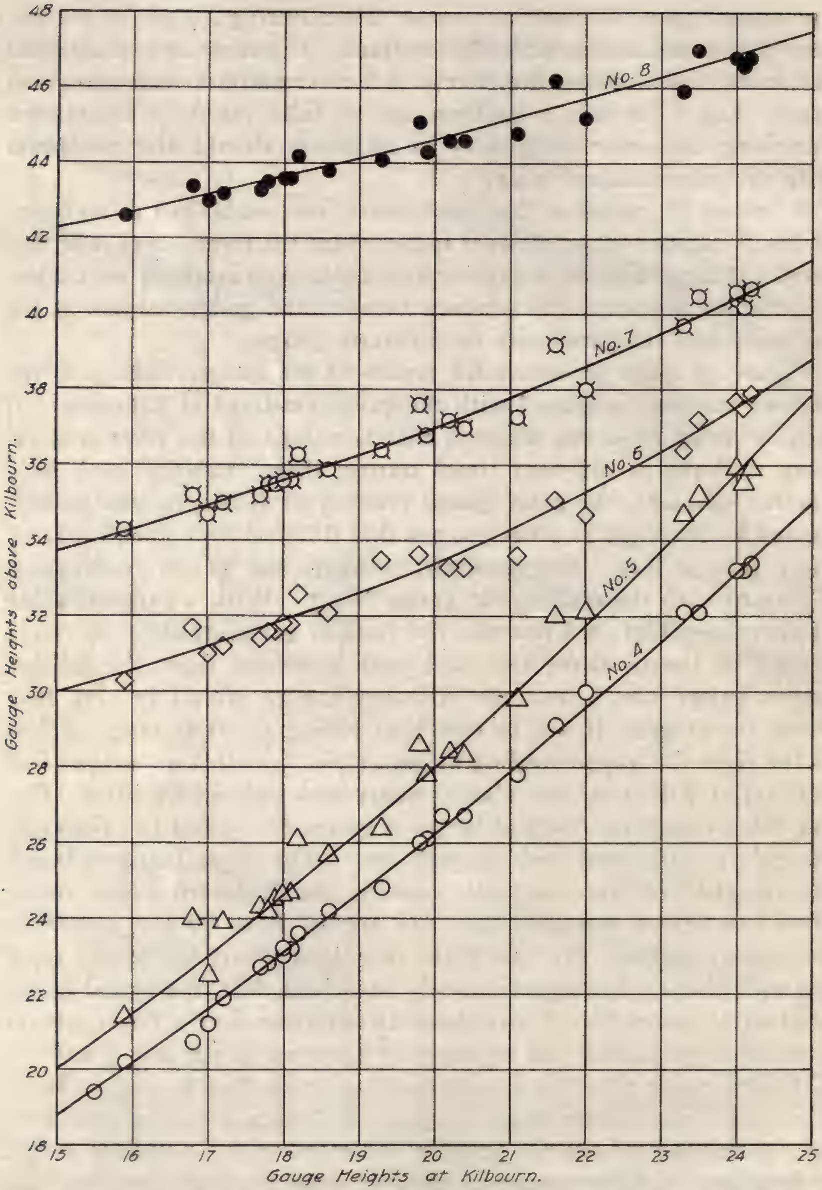

Fig. 48.—Relations of Gauge Heights at Various Stations on the Wisconsin River (see page 89).

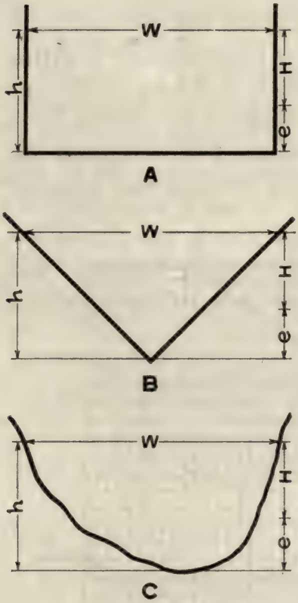

Fig. 49.

accurately representing the quantity of water flowing by the Kilbourn section. The above example of the variation in slope between the Kilbourn gauge and gauge No. 5 indicated practically the maximum abnormal conditions. The actual variation in flow at Kilbourn during these conditions was not determined and is not definitely known.

48. Relation of Gauge Heights to Flow.—The area of any cross-section equals the product of the height of the section into some function of its width:

$$a = h \times f(w) \tag{33}$$

In a rectangular cross-section $f = 1$ (see A, Fig. 49). In a triangular section $f = .5$ (see B, Fig. 49). In all cases of regular sections f can be mathematically expressed, and for irregular sections (see C, Fig. 49) the relation may be obtained by measurement. If the height of the surface is referred to a gauge height H, the zero of the gauge may or may not correspond with the bottom of the channel. If $H =$ the gauge height, then $h = H + e$, in which e is the distance from the bottom of the channel to the bottom of the gauge. Substituting, therefore, the value of h in equation (33) it becomes:

$$a = (H + e) \times f(w) = Hf(w) + ef(w), \tag{34}$$

And substituting this value in equation (11) it becomes:

$$Q = Hf(w)\, c \sqrt{rs} + ef(w)\, c \sqrt{rs} \tag{35}$$

With this equation, and with the flow in a fixed and uniform channel, if the relation can be established between r, s, c, e, w and f for each gauge height H, the corresponding value of Q can be determined. As these relations are mathematically expressed for uniform flow by the above equation, they can also be represented graphically by a curve which will show the relation between Q and H for all conditions of uniform flow that obtain in the given channel. Such a curve is called a discharge or rating curve. This equation (35) can be readily solved when f is a regular variable and when c, r and s can be determined. Where the function f, is an irregular

variable, no mathematical solution is practicable but the relations may be determined experimentally and can be expressed by a rating table or graphically by a rating curve. Such a rating table and curve can be constructed for every fixed channel or section of a stream for condition of uniform flow, no matter how irregular the section or how the values of the function of the section may vary for different gauge heights.

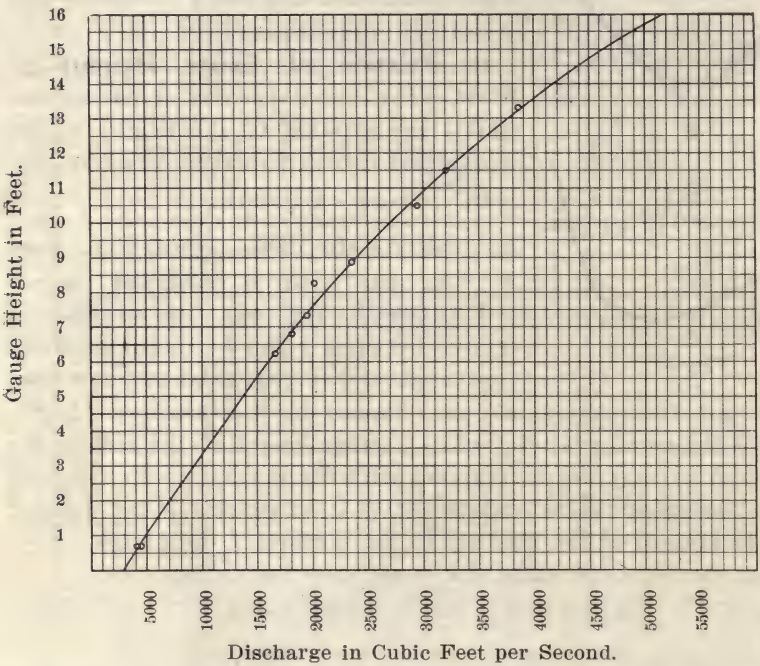

Fig. 50.—Rating Curve for Wisconsin River at Kilbourn, Wis.

Figure 50 shows a rating curve established for the Wisconsin River at Kilbourn, Wis. The small circles show the flow relative to gauge height at the time the observations were made. They were carefully made in a quite satisfactory section and fall fairly well on a smooth curve drawn from this data to represent the relation of gauge height to flow at similar or intervening heights.

The character of the rating curve for regular and irregular sections is shown by Fig. 51, page 93. Whenever the section remains similar for different gauge heights, the rating curve will be a smooth curve (see A, Fig. 51), but when irregularities occur in the section,

the curve becomes broken more or less according to the extent of the irregularity.

The form of the rating curve varies with the various conditions of the cross-section both at the immediate point and for a considerable distance above and below the location considered and can usually be determined only by detail observations. If, on account of overflow conditions, or sudden enlargements, the cross-section varies radically in form or if variations in roughness occur at a given height, then at this elevation a radical change in the slope of the rating curve is likely to occur (see B and C, Fig. 51).

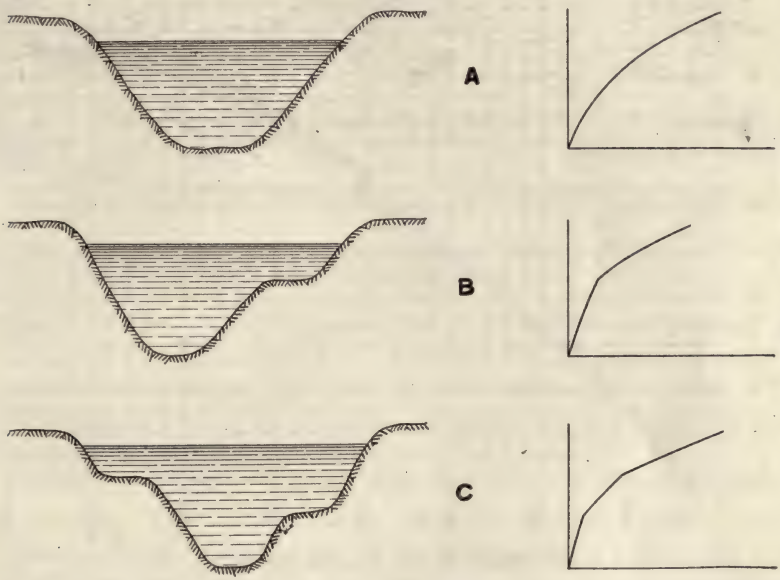

Fig. 51.—The Influence of the Stream Cross-section on the Rating Curve.

Any change in the bed of the stream may, and frequently does, modify to a considerable extent the rating curve, which must be expected to vary under such conditions to an extent that depends on the variations that take place in the cross-section and elevation of the stream bed. Such variations, however, are not, as a rule, of great magnitude and consequently will not usually affect the head materially at a given point, and especially not during high water under which conditions slight changes in the section become insignificant.

In Fig. 52, which shows the rating curve of the Wisconsin River at Necedah, Wis., as determined at different times during the years 1903 and 1904, an extreme change of head of about six inches will be noted for ordinary flows. When the change in head is of sufficient importance to warrant the expense, the river channel may be so dredged out as to restore the original head when the reduction in head is occasioned by the filling of the section.

An example of the actual change in channel conditions that sometimes affects the relation of head and flow is illustrated by Fig. 53, page

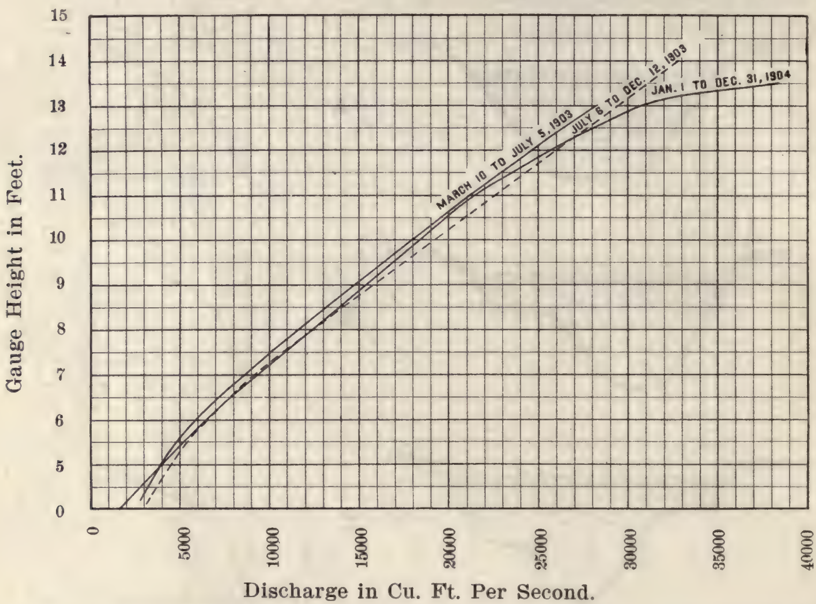

Fig. 52.—Rating Curves, Wisconsin River at Necedah, Wis., Showing Changes in Head Due to Changes in Cross-section.

95, which shows the changes that have actually taken place in the cross-section of the Missouri River near Omaha, Nebraska.

49. Variations in Velocity in the Cross-section of a Stream.—The velocity of flow of a stream varies greatly at different points in any cross-section. In any channel the friction of the sides and bed reduces the velocity of that portion of the stream in contact and adjacent to them. If the bed at different points of the cross-section is not uniform, as is always the case in the beds of natural streams, the retarding effects on different portions of the stream varies, and

a consequent variation in velocity results. The distribution of the velocities in the cross-section of the St. Clair River is shown in Fig. 54, page 96, by lines of equal velocity as actually measured. In this figure the effect of the friction of the bed and banks

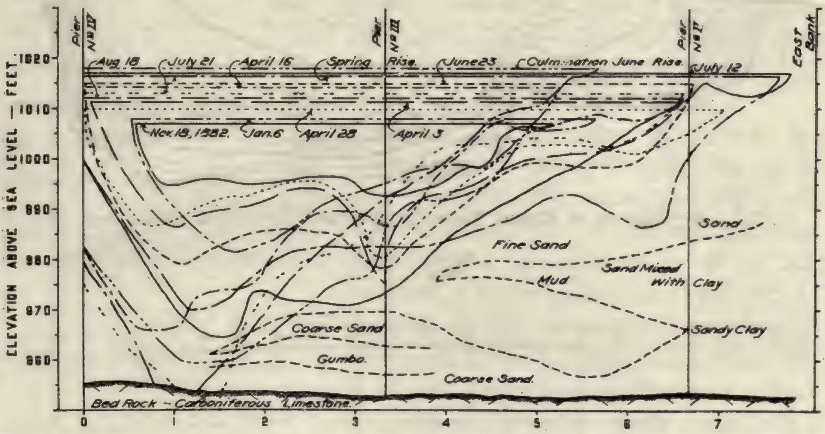

Fig. 53.—Variations in the Cross-section of the Missouri River near Omaha, Neb.* (see page 94).

is clearly shown. The friction between the stream surface and the atmosphere is also shown by the fact that the maximum velocity is not at the surface but is a short distance below the surface. The surface velocity may be modified radically by the direction and velocity of the wind.

Figure 55, page 96, shows the transverse curve of mean velocities in this section. The distribution of velocities in each vertical section is shown in Fig. 56, page 96. The velocities here shown are relative only as compared with each vertical. The velocity at the bottom of each curve is that shown in Fig. 54, page 96.

The distribution of velocities in an section is not the same under all conditions of flow but differs materially with the stage of the river. This is illustrated by Fig. 57, page 97, in which is shown three sections of the same stream illustrating conditions of low, medium and high water. Above each section is shown a corresponding transverse curve of mean velocities of flow. The change in the distribution of velocities as the stream increases should be noted.

The distribution of velocity is also affected by bends in the stream above the point of observation which tends to throw the current of

* Todd. Bull. 158 U. S. Geol. Surv.

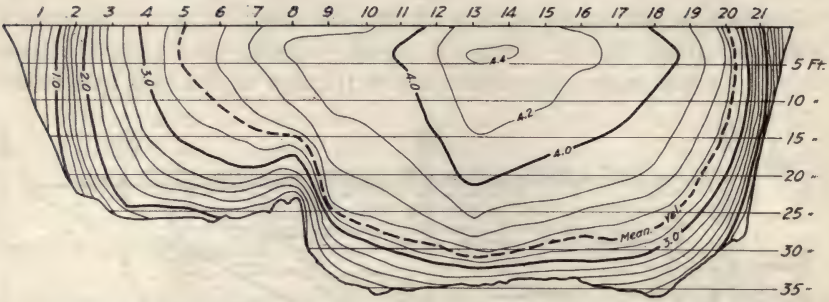

Fig. 54.—Curves of Equal Velocity, Section Dry Dock (see page 95).

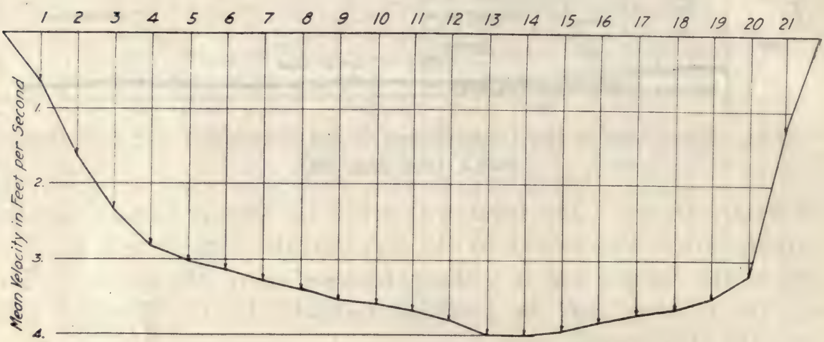

Fig. 55.—Transverse Curve of Mean Velocities, Section Dry Dock (see page 95).

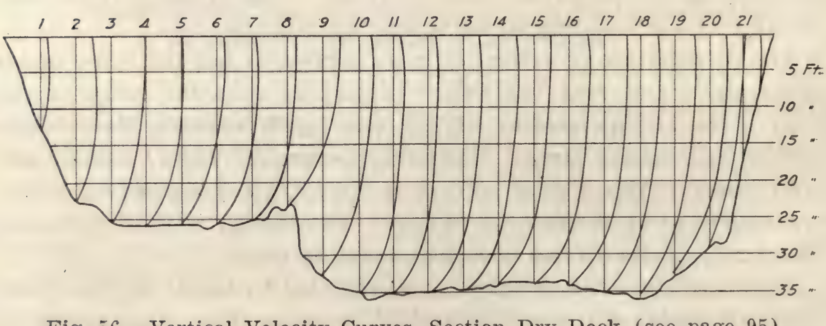

Fig. 56.—Vertical Velocity Curves, Section Dry Dock (see page 95).

the stream toward the concave side, and to cause a transverse slope in the section of the stream at the curve. Such a condition (see Fig. 58, page 98) creates cross currents and eddies and produces conditions of variable flow.

From Fig. 54, page 96, it will be seen that in any vertical line in a given section, the velocities will vary with the condition of the bed, and the influence of air current or ice at the surface. These conditions have an influence on the velocities in each section considered. Variations in the vertical velocities can be better studied by means of the vertical velocity curve, which can be obtained by means of velocity observations taken in a vertical line from the surface of the bed of the stream. Ideal curves under various conditions are illustrated by Fig. 59, page 98. Figs. 60, page 98, 61, page 99 and 62, page 100, are reproduced from the report of the State Engineer of New York for the year 1902. These diagrams show comparisons between the mean vertical velocities of streams having different classes of beds. From these illustrations it will be noted that there is a general similarity between the various velocity curves which aids materially in the measurement of stream flow. It will be noted, for example, that the mean velocity, in any vertical velocity curve from an open channel, lies near the point of six-tenths total depth but that with varying conditions this position may vary from fifty-five per cent. to about seventy-five per cent. of the depth. The velocity at six-tenths depth is found to

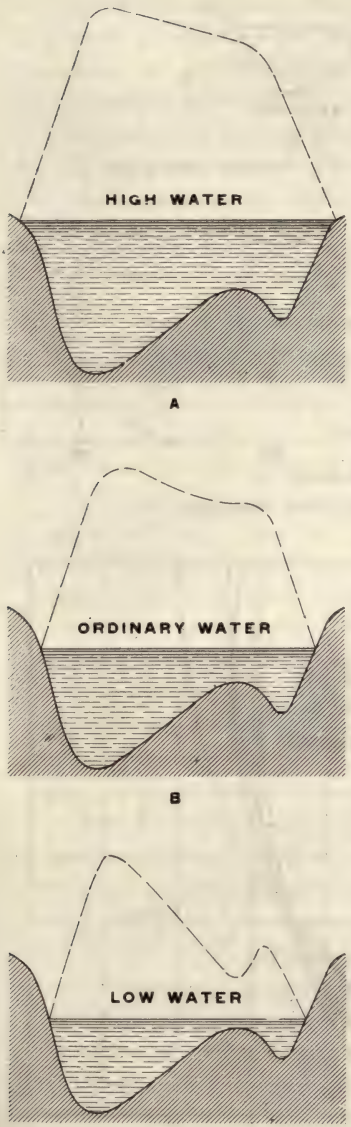

Fig. 57.

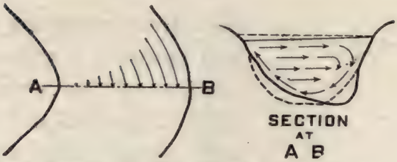

Fig. 58.—Effect of Bends on the Flow of a Stream (see page 97).

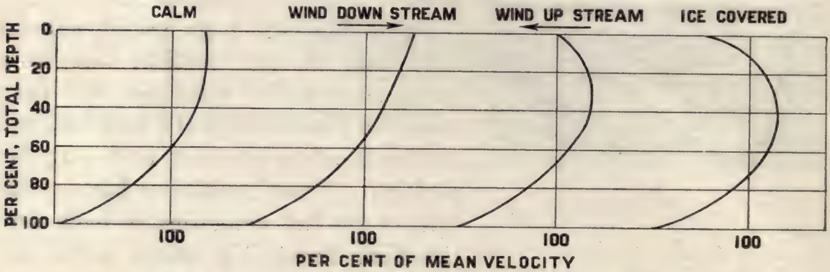

Fig. 59.—Ideal Vertical Velocity Curves (see page 97).

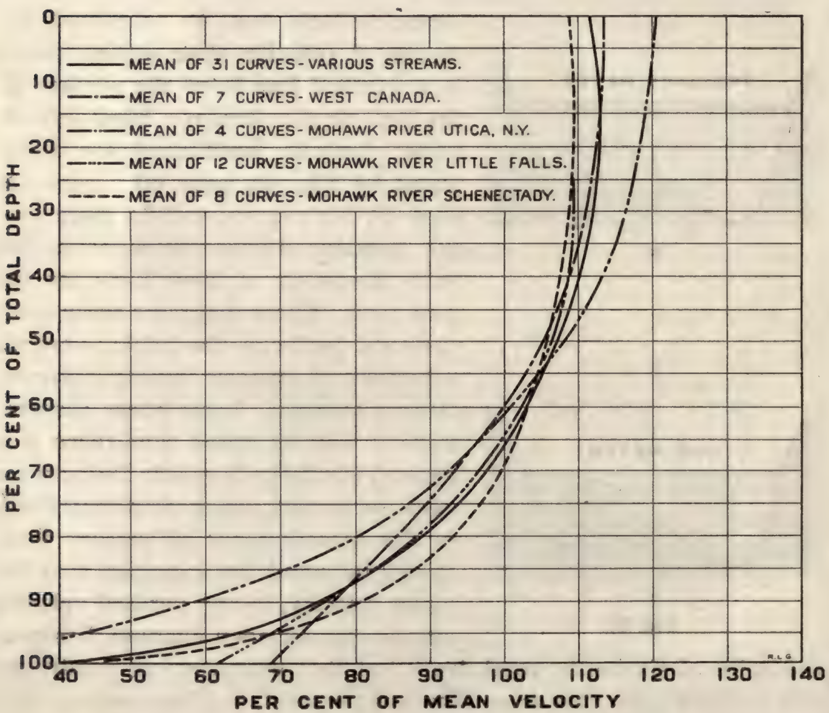

Fig. 60.—Mean Vertical Velocity Curves (see page 97).

average nearly 100 per cent. of the mean velocity, but may actually vary from ninety-five per cent. to 105 per cent. of the mean velocity. The velocity at the surface is subject to the external influence of atmospheric currents and is not so constant in its relation to the mean velocity. The surface velocity will average about 110 per cent. of the mean velocity of the vertical curve, but is found to vary with the variations in conditions from 105 per cent. to 130 per cent. of such velocity.

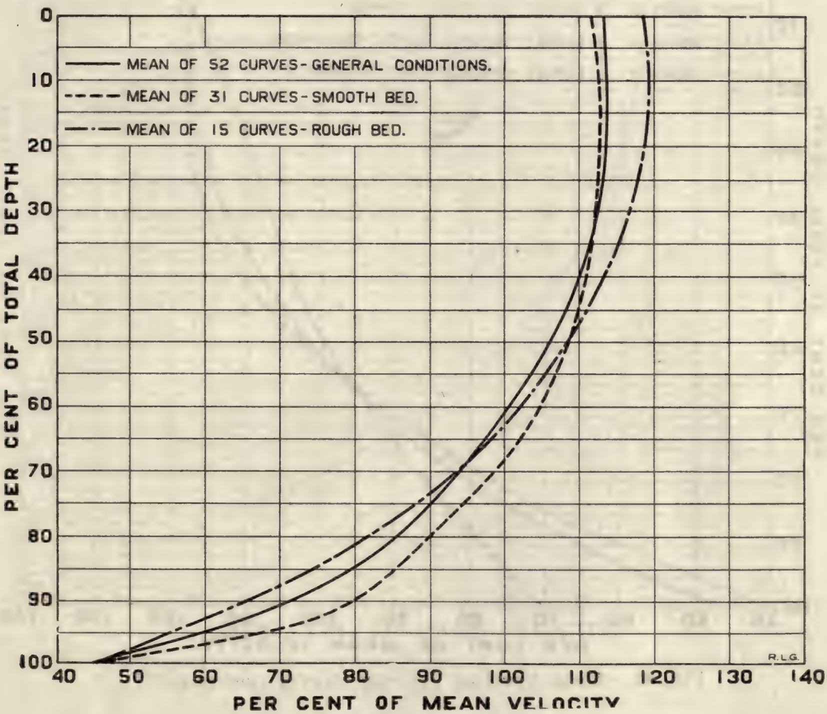

Fig. 61.—Mean Vertical Velocity Curves (see page 97).

50. **Effects of Ice Covering on the Distribution of Velocities.**— The effect of the formation of an ice sheet over a stream is to materially increase the surface friction and results in a rearrangement of velocities in the cross-section. As the ice sheets form in winter, the conditions will vary from that of an open stream to that of a closed channel. The velocities are gradually affected as the ice begins to form, until the entire surface is affected where the stream

becomes entirely covered. As the ice sheet thickens, more of the cross-section of the stream is occupied by the ice sheet, and greater friction results. Fig. 63, page 101, shows two vertical velocity curves, one for an open and one for an ice-covered channel. These may be regarded as typical of open and closed conditions between which the actual velocities will vary with the conditions of the ice.

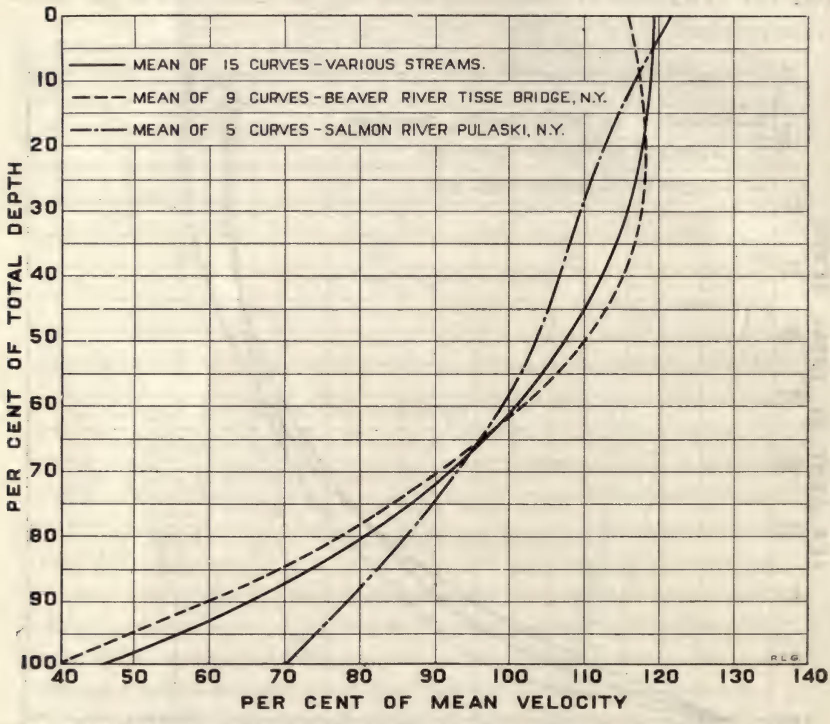

Fig. 62.—Mean Vertical Velocity Curves (see page 97).

The change in the distribution of velocities results in an entire change in the relation between gauge height and flow so that the rating curve for an open section will not apply to the river under ice conditions.

If therefore the stream flow is to be accurately determined during such condition, it becames necessary to establish the new relation between gauge height and flow.

As before noted, such relations vary somewhat with the conditions of the ice sheet but may be regarded as quite constant when

the section is fairly clear and deep. The relations between the rating curves for this open channel and for ice conditions as determined by the United States Geological Survey for the Wallkill River at Neupaltz, New York, is shown in Fig. 64, page 102.

Table 8, from an article by F. A. Tillinghast (see Engineering News, May 11th, 1905), shows the relations of maximum and mean velocities

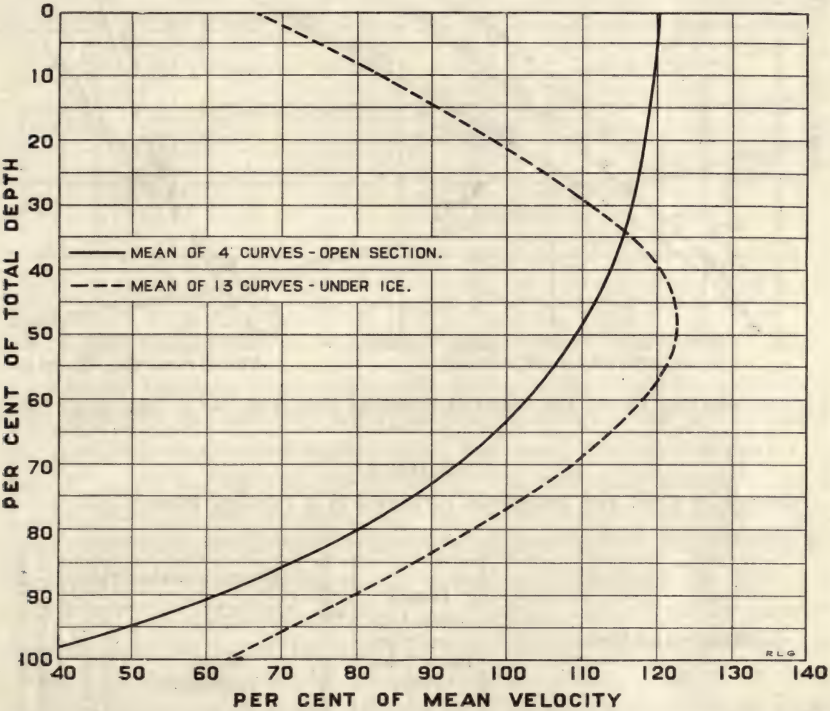

Fig. 63.—Comparative Mean Vertical Velocity Curves for Open and Ice Covered Section (see page 100).

in the verticals. It should be noted that there are two points of mean velocity under ice conditions that average eleven per cent. and seventy-one per cent. of the total depth below the surface. The point of maximum velocity is at an average depth of thirty-six per cent. of the total depth of the stream and averages 119 per cent. of the mean velocity.

51. The Back Water Curve.—The erection of a dam across a stream raises the surface at the point of construction an amount which depends on the height of the dam, the amount of water flowing in the stream, the amount of water diverted above the dam or

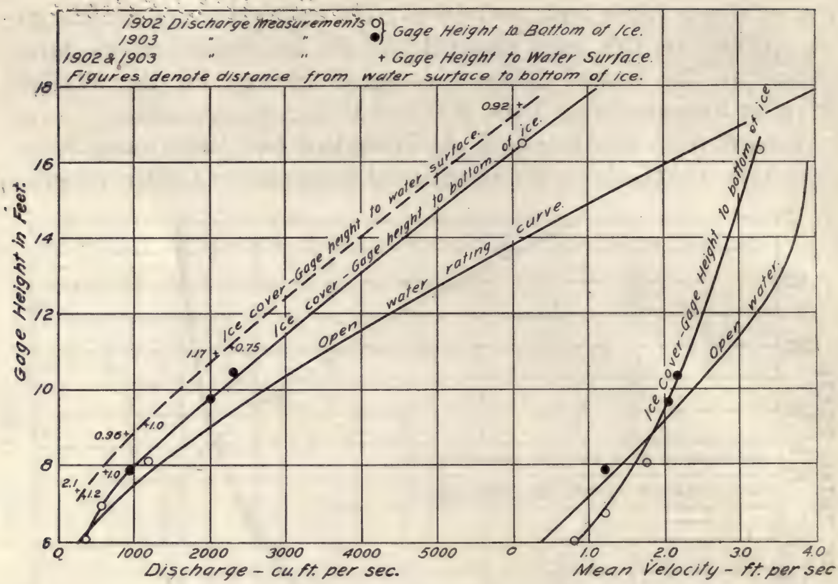

Fig. 64.—Rating Curve for Wallkill River at Neupaltz, N. Y. (see page 101).

TABLE 8.

Position of Mean and Maximum Velocities in a Vertical Plane Under Ice.

Stream and Place.	Depth from Under Surface of Ice Feet.	Number of Curves.	Proportionate Depth: Mean Velocity.		Proportionate Depth: Maximum Velocity.	Coefficient to reduce Max. to Mean
Wallkill at Neupaltz, N. Y.. (a)	4 to 12	20	0.12	0.71	0.38	0.85
Wallkill at Neupaltz, N. Y.. (b)	4 to 19	26	0.13	0.74	0.38	0.86
Esopus at Kingston, N. Y.. (a)	2.3 to 7.4	16	0.08	0.68	0.36	0.80
Esopus at Kingston, N. Y... (b)	5 to 8	8	0.11	0.73	0.37	0.85
Rondout at Rosendale, N. Y. (a)	4 to 8	5	0.08	0.68	0.35	0.82
Rondout at Rosendale, N. Y. (b)	5 to 7	8	0.13	0.21	0.35	0.86
Connecticut at Orford, N. H. (c)	2.5 to 7.7	18	0.11	0.69	0.35	0.85
Mean	....	....	0.11	0.71	0.36	0.84

Notes: a. By F. H. Tillinghast.
b. By W. W. Schlecht.
c. By C. A. Holden.

through water wheels, and the length and section of its spillway or the character and condition of its waste or flood gates, over or through which the waste water must pass.

The new elevation of the water surface under assumed conditions and with a given flow q, can be accurately determined from the assumption of the proposed design and the increased elevation of the water surface at the dam will therefore be known. The increase in depth and cross-section of the channel immediately above the

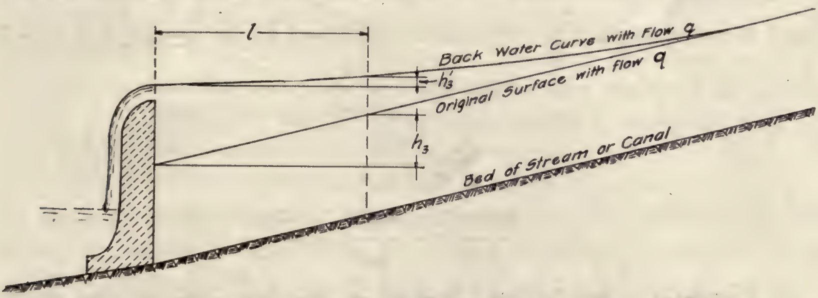

Fig. 65.—Relation of the Back Water Curve to Normal Flow.

dam under the new conditions, will decrease the velocity of flow v, and therefore the slope s will decrease, and hence at points above the dam the new water surface will approach nearer and nearer to the old water surface, as the distance from the dam increases, until finally the same will coincide, and the interference will cease. The new water surface is called the "back water curve," and the determination of the elevation of the curve and the point at which it "runs out" or coincides with the old water surface under various conditions of flow and especially under flood conditions, becomes an important problem in the determination of the amount of land needed for flowage or the amount of damage occasioned to property above the dam from the back water effect.

Most of the formulas for the back water curve are based on assumptions that are entirely incorrect, for all except the special channels to which they apply, and therefore cannot be applied to the irregular channels of streams. The approximate back water curve can best be calculated by the use of the simple formulas for flow in channels. From equation (9), page 72.

$$\text{(9)} \qquad h_3 = c' \frac{v^2 pl}{a} \text{ or } v = c \sqrt{\frac{ah_3}{pl}}$$

and from equation (11), page 73

$$\text{(11)} \qquad q = av$$

From equations (9) and (11)

(36) $$q^2 = a^2v^2 = \frac{c^2h_3a^3}{pl}$$

and

(37) $$h_3 = \frac{q^2pl}{c^2a^3} = \frac{q^2l}{c^2} \times \frac{p}{a^3}$$

52. **Conditions for Small Changes in Section.**—When the term $\frac{q^2l}{c^2}$ remains constant as it will approximately, if the change in sections is not too great for the two conditions to be considered, then:

$$h_3 : h'_3 :: \frac{p}{a^3} : \frac{p'}{a'^3}$$

or

(38) $$\frac{h_3}{h'_3} = \frac{p/a^3}{p'/a'^3}$$

and, therefore,

(39) $$h'_3 = \frac{h_3p'/a'^3}{p/a^3} = \frac{h_3p'a^3}{pa'^3} = \frac{h_3a^2r}{a'^2r'}$$

53. **Conditions for Large Changes in Section.**—When the depth of the new section changes greatly, in relation to the first section, q and l will remain constant but c will vary, hence under these conditions

$$h_3 : h'_3 = \frac{p}{c^2a^3} : \frac{p}{c'^2a'^3}$$

and

(40) $$h'_3 = \frac{h_3p'a^3c^2}{pa'^3c'^2} = \frac{h_3a^2rc^2}{a'^2r'c'^2}$$

In the practical solution of this problem, the various sections of the stream considered, are determined by surveys and the relations of a, h_3, p and r are determined or accurately calculated from such surveys for the various reaches between sections under various volumes q, of flow, the various values of q being at the same time determined from meter measurement and the corresponding values of c calculated.

The effect of the proposed change on the first or lower cross-section must first be calculated and that on the other sections of the reaches can first be estimated or assumed, and with such assumptions as a basis, the corresponding average values of p', a' and c' can be determined. Then from these and the known values of h_3, p, a, and c, the value of h'_3 can be calculated by use of equation 40. The results of this calculation will probably show that the assumed section is incorrect and new values for the properties of

the section may be assumed with the preliminary value of h'_3 so obtained as a guide, and the computations repeated until the assumed value of h'_3 and the computed value check within reasonable limits.

Then, having thus determined the value for h'_3 at the new section at the upper end of the first reach, the total depth is known approximately, and consequently the other properties for this cross-section of the stream are known from the results of the survey. The slope or fall in the second reach may then be computed in the same manner using the properties of the new section as a basis, and so on until the water surface under the new order of flow practically coincides with the original water surface, showing that the limit of the back water effect has been reached.

In the practical application of this method of predicting the form of the surface curve under the changed conditions of flow, the surveys first made to determine the actual existing conditions should cover a range as wide as possible, that is to say, the observations should include data for stages of the stream varying as widely as can be obtained between extreme low and extreme high water conditions and should be taken at the several sections of the stream which may be selected. The value of c for the various reaches of the stream between the points of observation for the various stages of the flow are then computed from the formula

$$q = ca\sqrt{\frac{ah_3}{pl}} \qquad (11)$$

These values may then be platted against the values of r at each section and the curve of variation in the value so found may be extended and thus serve as a basis for estimating the value to be used in the computations of the back water curve under the new conditions which will be caused by the proposed obstruction.

It is well, in obtaining the data for the existing conditions of flow, to include a number of sections in each reach considered, and in this manner obtain a value for c more nearly equal to the real average value than would be given by the computation based on sections only at the ends of the reach in case any considerable length is included in one reach. The principal error in all such calculations where the surveys are so complete that a, p and r are accurately known lies in the variations in c as the water surfaces rise or fall from the section for which these values of c have been actually determined.

54. **Back Water Computation.**—A method of computing the back water curve which applies more particularly to regular channels, but which may be used to obtain a rough approximation for natural water courses in the absence of sufficient data to use the foregoing method, is as follows: The values of c to be used in the Chezy formula may be obtained by Bazin's formula (see Fig. 41, page 75).

$$c = \frac{87}{.552 + \frac{m}{\sqrt{r}}}$$

and this value used in the equation for flow $v = c\sqrt{rs} = c\sqrt{\frac{ah_3}{pl}}$ we have:

(41) $$h_3 = \frac{v^2pl}{c^2a} = \frac{v^2l}{c^2r}$$

The portion of the stream under consideration is to be divided into reaches and the drop of the water surface in each reach is computed.

A depth at the upper end of the first reach must be assumed, the value of v, a, c and r computed and averaged with values taken at other points throughout the reach and these average values used in the equations.

Following is a problem computed by this method for assumed conditions. It should be borne in mind that in the practical application of this method, the coefficient c does not necessarily vary as given by either Bazin's or Kutter's Formulas, and in fact is quite apt to follow an entirely different law. In this connection attention is called to Fig. 44, page 82, as showing the variations actually obtained from observations on the Wisconsin River.

PROBLEM

Consider an earth canal or ditch with trapezoidal section, base width 20.0 feet, depth 5 feet, side slopes 1½ to 1. Slope one foot in 2000 or gradient .0005. Consider a dam with sharp crest raises the water surface 2.0 feet. Compute the back water curve by finding the elevation of water surface at each thousand feet upstream from the obstruction.

Original section area = 137.5 sq. ft.
Wetted perimeter = 38.02 feet.
Hydraulic radius = 3.615 feet.

$$c = \frac{1.30}{.552 + \frac{1.30}{\sqrt{3.615}}} = 70.4$$

$v = c\sqrt{rs} = 2.99$ feet per second.

$q = av = 411.4$ second feet.

This value of q will remain constant throughout the computation. When the water surface is raised 2.0 feet by the obstruction, the new depth immediately above the obstruction will be 7.0 feet. Then for this particular point:

$$a' = 213.5$$
$$p' = 45.24$$
$$r' = 4.72$$
$$v' = \frac{q}{a} = 1.93$$

Now considering the point 1000 feet from the obstruction. Assume the depth will be 6.65 feet, then the average properties of the lower and upper sections will be

$$a' = 206.45 \text{ sq. ft.}$$
$$v' = 1.99$$
$$r' = 4.64$$
$$c' = 75.0$$

Using these values in Equation (41)

$$h'_3 = \frac{\overline{1.99^2} \times 1000}{\overline{75^2} \times 4.64} = .152$$

Let the elevation of canal bed at obstruction = 00.0
Then elevation of water surface at obstruction = 7.0
The elevation of water surface at 1000 feet = 7.152
The elevation of canal bed at 1000 feet = 0.50

Depth of water at 1000 feet = 6.652

Now consider a point 2000 feet from the obstruction. Assume the depth will be 6.34 then the average value of the properties of the second reach will be

$$a' = 193.36 \text{ sq. ft.}$$
$$v' = 2.13 \text{ ft. per sec.}$$
$$r' = 4.450$$
$$c' = 74.4$$

$$h'_3 = \frac{\overline{2.13^2} \times 1000}{\overline{74.4^2} \times 4.45} = .184$$

Elevation water surface at 1000 feet.......................... 7.152
Fall in second reach.. .184

Elevation water surface at 2000 feet.......................... 7.336
Elevation canal bottom at 2000 feet........................... 1.000

Depth of water at 2000 feet................................... 6.336

Consider a point 3000 feet from obstruction. Assume the depth at this point will be 6.04 feet, then the average values of the channel properties will be:

$$a' = 181.27 \text{ sq. ft.}$$
$$v' = 2.27 \text{ ft. per sec.}$$
$$r' = 4.275$$
$$c' = 73.6$$

And
$$h'_3 = \frac{\overline{2.27}^2 \times 1000}{\overline{73.6}^2 \times 4.275} = .222$$

Elevation of water surface at 2000 feet	7.336
Fall in third reach	.222
Elevation water surface at 3000 feet	7.558
Elevation canal bottom at 3000 feet	1.500
Depth of water at 3000 feet	6.058

In general more than one assumption as to the depth must be made before one will be found which will check with the computed value.

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CHAPTER V

THE MEASUREMENT OF STREAM FLOW

55. Letters and Symbols.—The letters and symbols used in this chapter have the following significance:

h = Total head (feet).
h' = Velocity head (feet).
v = Velocity of flow (feet per second).
v' = Velocity of approach to weir (feet per second).
q = Discharge (cubic feet per second).
q' = Discharge through a single section of stream cross-section.
q'' = Discharge through a double section of stream cross-section.
q_1 = Discharge of stream as deterimned by float measurement.
m = Coefficient for orifices under low heads.
c = Coefficient of discharge.
n = Number of end contractions of a weir.
L or l = Length in feet of weir crest, section of stream width, etc.

$$D = \text{Ratio}\ \frac{\text{distance from bottom of float to bottom of stream}}{\text{depth of stream.}}$$

d_0, d_1, d_2 – – – = Consecutive depths across a stream.
d''_m = Mean depth for double section across a stream.
d'_m = Mean depth for a single section across a stream.
v_0, v_1, v_2 – – – – = mean velocities in the verticals where the depths are d_0, d_1, d_2 – – – respectively.
v''_m = Mean velocity for a double section of stream cross-section.
v'_m = Mean velocity for a single section of stream cross-section.
g = Acceleration due to gravity (32.2 feet per second).

56. Necessity for Stream Flow Measurements.—In order to ascertain the value of a stream for water power purposes, it is necessary to determine the amount and variations in its continuous flow either by comparison with the flow of other streams or by the direct observation of the flow of the stream itself. As has already been shown, the latter method is by far the most satisfactory as the determination of the actual flow of the stream eliminates all errors of comparison and the necessity for any allowances or modifications on account of differences in geological, geographical, topographical or meteorological conditions on the drainage area.

The Hydrographic Division of the United States Geological Survey has undertaken the gauging of a large number of streams in the

United States and has established numerous gauging stations at which observations have been made for a number of years. This data, references to which are given in the list of literature appended to Chapter VI, is of great value for comparative purposes. It is seldom, however, that, when a stream is to be investigated for water power purposes, flow data, at the particular point under consideration, is available. One of the first duties of the engineer, therefore, usually consists in making measurements of the stream flow and establishing stations at which the daily flow can be observed and recorded.

The methods in use by the United States Geological Survey are the result of much study and investigation and probably represent the most practical methods for making such observations with a fair degree of accuracy. Many of the methods and suggestions in this chapter are based on the methods and conclusions of the Survey as modified by the experience and practice of the writer.*

57. Methods for the Estimate or Determination of Flow in Open Channels.—There are three general methods of estimating or determining the flow of water in streams with open channels.

First: By the measurement of the cross-section and slope and the calculation of flow by Chezy's formula, together with Kutter's or Bazin's formulas for estimating the values of the coefficient.

Second: By means of orifices, overflow weirs or dams of such form that the coefficient of discharge is known.

Third: By the measurement of the cross-section area and the velocity of current passing through the same.

The method which should be selected for any particular location depends on the physical conditions of the problem, the degree of accuracy required, the expense which may be permissible and the length of time during which the record is to be continued.

58. Estimates from Cross-section and Slope.—Chezy's formula,

$$v = c\sqrt{rs} \tag{10}$$

together with the formulas of Kutter and Bazin, for the determination of the flow of streams, has already been discussed in Chapter IV. Much information is now available in regard to the value

* These methods are described in detail in Water Supply and Irrigation Papers No. 94, entitled, "Hydrographic Manual of the United States Geological Survey," and No. 95, entitled "Accuracy of Stream Measurements." See also "River Discharge" by J. C. Hoyt and N. C. Grover,—John Wiley and Sons, 1907.

of the coefficient c, but this value varies greatly in different streams, in accordance with the conditions of the beds, and in the same stream under various volumes of flow. The results obtained from the application of these formulas are therefore necessarily very approximate. The method, however, is of considerable value in estimating the flood discharge of streams and in obtaining an approximate knowledge of flow under other conditions where other methods are not available or are difficult of application.

In using this method two or more cross-sections of the stream should be measured on reaches of the river where the cross-section and other conditions are fairly uniform and can be readily determined and at a time when the flow is steady. It is also important that the stream in which the flow is to be estimated shall be comparable in cross-section, depth, and other conditions, on which the value of the coefficient c depends, with other streams on which the value of c has been determined.

59. The Use of Orifices and Weirs for Discharge Measurements.— The gates or openings in permanent dams and the overflow section of same, when of such form that the coefficients of discharge are known or can be closely approximated, and temporary dams with openings or weirs of desired form and known coefficients, constructed for the purpose of measurement, afford the best practical method for measuring stream flow.

Orifices may be considered as free or submerged. The discharge of submerged weirs has not been determined with sufficient accuracy for gauging purposes.

In order to assure accurate results in weir measurements, the following conditions must be fulfilled:

First: The dam or weir must have sufficient height so that back water will not interfere with the free fall over the same.

Second: The dam or weir body must be so constructed that no leak of appreciable size will occur during the time when it is utilized for measuring purposes.

Third: The abutments of the dam or sides of the weir must be so constructed as to confine the flow over the dam at all stages: otherwise the weir will be useless for measurements during flood conditions.

Fourth: The crest of the weir must be level and must be kept free from obstructions caused by floating logs or ice.

Fifth: The crest of the dam or weir must be of a type for which

coefficients for use in the ordinary weir formula have been determined * (see Appendix C, page 814.)

Sixth: If the dam has an adjustable crest, great care must be used to prevent leakage along such crest and to keep a complete and detailed record of the condition of the crest during the time of the observations.

Seventh: If water is diverted around the dam, which is usually the case when a dam is built for power purposes or for navigation, the diverted water must be measured or estimated and added to the amount passing over the dam. Such diverted water can sometimes be measured by a weir or current meter. When such water is used in water wheels, an accurate record of the gate opening of the wheels can be kept, from which the amount of water used in the wheels can be estimated if the wheel's discharge has been calibrated of if the wheel is of some well known type.† The conditions for the accurate determination of weir discharge should be such as not to involve the use of low heads of less than six inches over broad crested dams.

Measurements by means of a weir or dam have the general advantage of continuity of record during the periods of ice and flood and the disadvantage of uncertainty of the coefficient to be used in the weir formula, of complication by the diversion of water around the dam, and the interference of flow by the occasional lodgement of material, or of injury to the crest.

60. The Flow of Water Through Orifices.—It is found that water flowing through an orifice in the side of a vessel acquires a velocity practically equal to that which would be acquired by a falling body in passing through a space equal to the head above the center of the opening, i. e.,

$$v = \sqrt{2gh} = 8.025\sqrt{h} \tag{42}$$

in which

v = velocity of spouting jet.
g = acceleration of gravity = 32.2.
h = head on opening.

The discharge through the opening would therefore be theoretically

$$q \propto va = a\sqrt{2gh} \tag{43}$$

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* See Water Supply and Irrigation Paper No. 200,—Weir Experiments, Coefficients and Formulas—by R. E. Horton.

† See Water Supply and Irrigation Paper No. 180,—Turbine Water Wheel Tests and Power Tables—by R. E. Horton.

or practically

$$(44) \qquad q = ca\sqrt{2gh}$$

where c is a coefficient varying with the size and shape of the orifice and with various other factors.

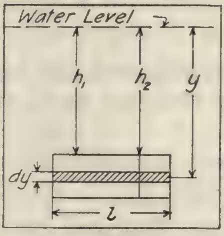

Fig. 66.

A more accurate determination of the theory of flow through a given orifice is derived as follows: If a thin opening is considered at a depth y below the surface, the discharge through the elementary section ldy would be

$$(45) \qquad dq = ldy\sqrt{2gy}$$

Integrating this equation between the limits h_2 and h_1 we obtain the following:

$$(46) \qquad q = \tfrac{2}{3}l(h_2^{\frac{3}{2}} - h_1^{\frac{3}{2}})\sqrt{2g} \text{ or practically}$$

$$(47) \qquad q = m\tfrac{2}{3}\, l\sqrt{2g}\,(h_2^{\frac{3}{2}} - h_1^{\frac{3}{2}})$$

m being the coefficient of practical modification due to condition of the orifice.

For all except very low heads, the equation

$$(44) \qquad q = ca\sqrt{2gh}$$

is sufficiently accurate for all practical purposes. For complete contraction (i. e. with a sharp edged orifice) the average value of the coefficient will be about .60. With rounded corners the discharge will be gradually increased. When the curve of entrance entirely eliminates the contraction, the value of the coefficient will approach unity. The values of the coefficient with large orifices and various conditions of the suppression of contraction are illustrated both for submerged orifices and short tubes in Table 9, taken from experiments by Mr. C. B. Stewart at the hydraulic laboratory of the University of Wisconsin.

The Forms of Entrance and Outlet Used for the Tubes in the Experiment Were as Follows:

A Entrance; all corners 90°.
Outlet; tube projecting into water on down stream side of bulkhead.

a Entrance; contraction suppressed on bottom.
Outlet; tube projecting into water on down stream side of bulkhead.

b Entrance; contraction suppressed on bottom and one side.
Outlet; tube projecting into water on down stream side of bulkhead.

c Entrance; contraction suppressed on bottom and two sides.
Outlet; tube projecting into water on down stream side of bulkhead.

d Entrance; contraction suppressed on bottom, two sides and top. Outlet; tube projecting into water on down stream side of bulkhead.

From this table it will be noted that a partial suppression of contraction does not always improve results, and that by complete suppression, the coefficient is greatly increased with a corresponding decrease in lost head.

TABLE 9.

Value of the Coefficient of Discharge for Flow Through Horizontal Submerged Tube, 4 Feet Square, for Various Lengths, Losses of Head and Forms of Entrance and Outlet.

Loss of head, h_3 in feet.	Forms of Entrance and Outlet.	Length of tube, in feet.						
		0.31	0.62	1.25	2.50	5.00	10 0	14.0
		Value of the coefficient, c.						
.05....	A	.631	.650	.672	.769	.807	.824	.838
	a	.762	....	....	.742	.810	....	.848
	b	.740	....	....	.769	.832	....	.862
	c	.834	....	....	.769	.875	....	.890
	d	.948	....	....	.943	.940	.927	.931
.10....	A	.611	.631	.647	.718	.763	.780	.795
	a	.636	....	....	.698	.771	....	.801
	b	.685	....	....	.718	.791	....	.813
	c	.772	....	....	.718	.828	....	.841
	d	.932	....	....	.911	.899	.892	.893
.15....	A	.609	.628	.644	.708	.758	.779	.794
	a	.630	....	....	.689	.767	....	.803
	b	.677	....	....	.708	.787	....	.814
	c	.765	....	....	.708	.828	....	.839
	d	.936	....	....	.910	.899	.893	.894
.20....	A	.609	.630	.647	.711	.768	.794	.809
	a	.632	....	....	.694	.777	....	.819
	b	.678	....	....	.711	.796	....	.833
	c	.771	....	....	.711	.838	....	.856
	d	.948	....	....	.923	.911	.906	.905
.25....	A	.610	.634	.652	.720	.782	.812	.828
	a	.634	....	....	.705	.790	....	......
	b	.683	....	....	.720	.809	....	......
	c	.779	....	....	.720	.854	....	......
	d	.965	....	....	.938	.928	....	......
.30....	A	.614	.639	.660	.731	.796	.832	.850
	a	.639	....	....	....	....	....	......
	b	.689	....	....	....	....	....	......
	c	.788	....	....	....	....	....	......
	d	.984	....	....	....	....	....	......

61. Flow Over Weirs.—In a weir $h_1 = o$ (see Fig. 66, page 115). Hence equation (47) becomes

$$q = m\,(\tfrac{2}{3})\,l\,\sqrt{2g}\,h^{\frac{3}{2}} \tag{48}$$

in which h is the head on the crest of the weir, i. e., the vertical distance from the water level above to the crest of the weir.

For practical use the coefficient m together with the constants $\frac{2}{3}$ and $2g$ are combined as follows:

$c = m\,\frac{2}{3}\,\sqrt{2g} = M\,\sqrt{2g}$ and equation (48) becomes

$$q = c\,lh^{\frac{3}{2}} \tag{49}$$

The value of m and consequently of c varies with the shape of the weir and with other factors and must be determined experimentally. This has been done with weirs of many forms, both by Bazin in France and by Rafter and Williams at the Cornell hydraulic laboratory. The results of these experimental determinations are given by Figs. 431 to 436, inclusive (see Appendix C, pages 814 to 820).

In practice many weir formulas are in use, based on various experiments and observations. The formula of Francis, equation (50), is probably the best known in this country. It is best adapted to long, sharp crested weirs without end contractions.

$$q = 3.33\,lh^{\frac{3}{2}} \tag{50}$$

A number of different formulas for the flow over weirs are given on Fig. 437, page 820, and the flow as calculated by these formulas is shown on the diagram. L in these formulas represents the length of the weir crest which in the dimension above is represented by l.

Figure 436, page 819, shows graphically the results of the application of the value of c as given in Figs. 431 to 435, pages 814 to 818, as compared with Francis' formula.

In small weirs the effect of end contraction and of the velocity of approach becomes important and corrections to the formulas must be applied in order to allow for those influences.

If n = the number of end contractions and the effect of each is to reduce the effective length of the weir by one-tenth the head on the weir, equation (49) will become

$$q = c\left(l - n\frac{h}{10}\right)h^{\frac{3}{2}} \tag{51}$$

The effect of the velocity of approach, for a given quantity, is to reduce the head on the weir by the velocity head. This reduction is given by the formula:

$$h' = \frac{v'^2}{2g} \tag{52}$$

in which v' = velocity of approach and h' = velocity head.

To allow for the influence of velocity of approach h' must be added to h and equation (51) becomes

$$q = c\left(1 - n\frac{h}{10}\right)(h + h^1)^{\frac{3}{2}} \tag{53}$$

Experimental results at the hydraulic laboratory of the University of Wisconsin show that for small sharp crested weirs, with end contraction, the formula (54) of Hamilton Smith, Jr., is practically correct:

$$q = c\,\tfrac{2}{3}\sqrt{2g}\,lh^{\frac{3}{2}} \tag{54}$$

In this formula

c = coefficient of discharge (to be taken from Table 10).

h = observed head on crest (H) plus correction due to velocity of approach.

Variations in the forms of the crest of weirs and in the arrangement of sides and bottom of the channel of approach cause considerable variation in their discharging capacity. It is therefore apparent that unless the conditions closely agree with those on which experimental data is available that the error of calculation may be considerable.

TABLE 10.

Coefficient of Discharge c for Use with Hamilton Smith, Jr.'s Formula (54) for Flow of Water Over Sharp Crested Weirs Having Full Contraction.

l = length of weir in feet.

Effective head = h	.66	1(?)	2	2.6	3	4	5	7	10	15	19
.1	.632	.639	.646	.650	.652	.653	.653	.654	.655	.655	.656
.15	.619	.625	.634	.637	.638	.639	.640	.640	.641	.642	.642
.2	.611	.618	.626	.629	.630	.631	.631	.632	.633	.634	.634
.25	.605	.612	.621	.623	.624	.625	.626	.627	.628	.628	.629
.3	.601	.608	.616	.618	.619	.621	.621	.623	.624	.624	.625
.4	.595	.601	.609	.612	.613	.614	.615	.617	.618	.619	.620
.5	.590	.596	.605	.607	.608	.610	.611	.613	.615	.616	.617
.6	.587	.593	.601	.604	.605	.607	.608	.611	.613	.614	.615
.7	.585	.590	.598	.601	.603	.604	.606	.609	.612	.613	.614
.8	......	.....	.595	.598	.600	.602	.604	.607	.611	.612	.613
.9	......	.....	.592	.596	.598	.600	.603	.606	.609	.611	.612
1.0	......	.....	.590	.593	.595	.598	.601	.604	.608	.610	.611
1.1	......	.....	.587	.591	.593	.596	.599	.603	.606	.609	.610
1.2	......	.....	.585	.589	.591	.594	.597	.601	.605	.608	.610
1.3	......	.....	.582	.586	.589	.592	.596	.599	.604	.607	.609
1.4	......	.....	.580	.584	.587	.590	.594	.598	.602	.606	.609
1.5	......	.....	.....	.582	.585	.589	.592	.596	.601	.605	.608
1.6	......	.....	.....	.580	.582	.587	.591	.595	.600	.604	.607
1.7	......	.....	.....	.....	.....	.....	.....	.594	.599	.603	.607
2.0	......	.....	.....	....	.....	.....	.....	.....	.....	.....	.....

62. Measurement of Flow by the Determination of Velocity.—The discharge of a stream, or the quantity of water flowing past a certain section of the stream in a given time, is the product of two factors: first, the area of the cross-section; and second, the mean velocity of flow through said section.

If the flow in the cross-section of the stream were uniform the measurement of the flow would be a simple matter. A surface float,

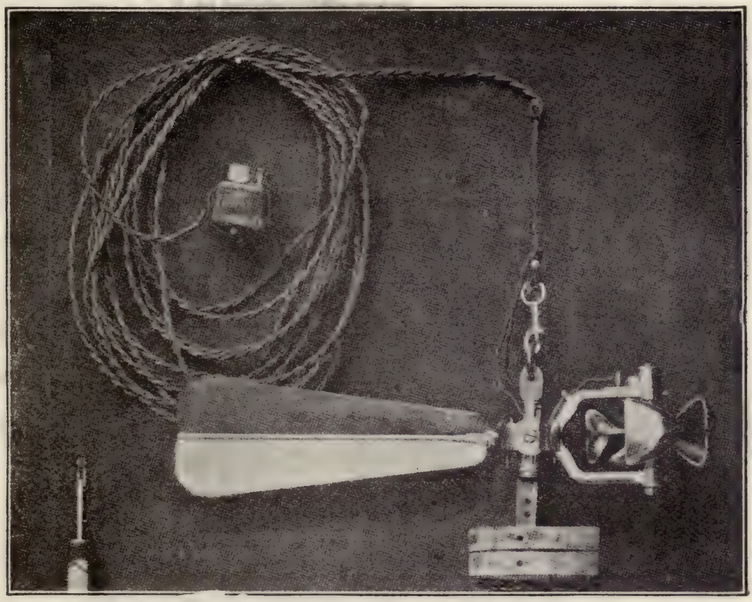

Fig. 67.—Price Electric Current Meter with Buzzer.

timed between given stations, or a current meter placed at any point in the cross-section, would then indicate the average velocity. Such conditions, however, never obtain. It is therefore necessary to ascertain the mean velocity of flow in the section which is a much more difficult matter.

Two methods of measuring the velocity of a stream are in use: first, by the use of a current meter, and second, by the use of floats. Each of these methods has advantages peculiar to itself, which must be known and appreciated in order that intelligent measurements may be made.

63. The Use of the Current Meter.—The current meter (Figs. 67 and 68) is an instrument designed to revolve freely with the current so that by determining the number of its revolutions the velocity of

the current will be known. A well made current meter carefully maintained and frequently rated is reasonably accurate when properly used under conditions to which it can be applied. As the friction of operation is rarely constant, the relation of current velocities to number of revolutions is not always strictly proportional and it is necessary to determine the relation between the revolutions of the

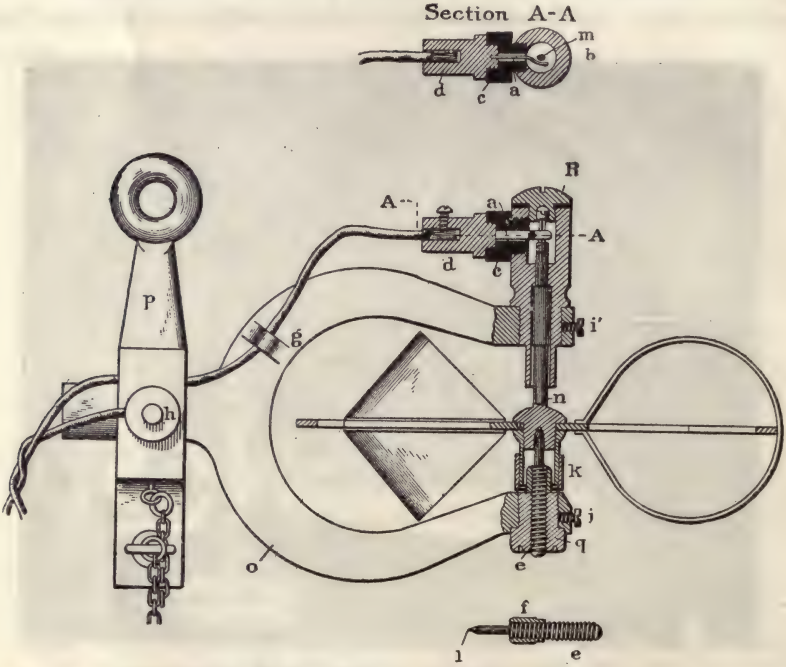

Fig. 68.—Cross-section of Small Price Electric Current Meter, Showing Details.*

meter and the corresponding velocity of water. This is accomplished by rating the meter, which is usually done by passing it through still water at known velocities and noting the results. It is assumed that the same relation will exist between the revolutions of the meter and its longitudinal velocity through still water and between its revolutions and the velocity of flowing water when the meter is held in a similar position in a stream. The meter should be rated under conditions as nearly similar as possible to those under which it was, or is to be, used. The meter when being rated is

* From Water Supply and Irrigation Paper No. 94,—Hydrographic Manual by E. C. Murphy, J. C. Hoyt and G. B. Hollister.

usually attached to some movable device (see Fig. 69) such as a carriage or boat which is propelled by hand or machinery at a known rate over a fixed distance. Observations of the revolutions of the meter at various rates of speed are noted and the relation is then established between the velocity of the meter and the revolutions of the meter wheel. This data may be platted upon cross-section paper

Fig. 69.—Current Meter Rating Station at Denver, Colo.*

or so arranged in tabular form that the corresponding velocity may be immediately ascertained when the revolutions of the meter are known (see Fig. 70, page 122). Experiments have shown that with velocities less than one-half of a foot per second little or no dependence can be placed upon the meter observations and that for velocities below one foot per second, the meter usually under registers. Where such low velocities obtain, float measurements are believed to be more accurate.

* From Hydrographic Manual.

64. Current Meter Observations and Computation.—On account of the great variation in velocity at different points in the cross-section, the flow through any unit of area may vary more or less from the flow through other similar areas. On this account it is desirable, in order to systematically survey the velocities in a cross-section, as well as for ease in calculation, to divide the cross-section area into parts, both horizontally and vertically, and determine the

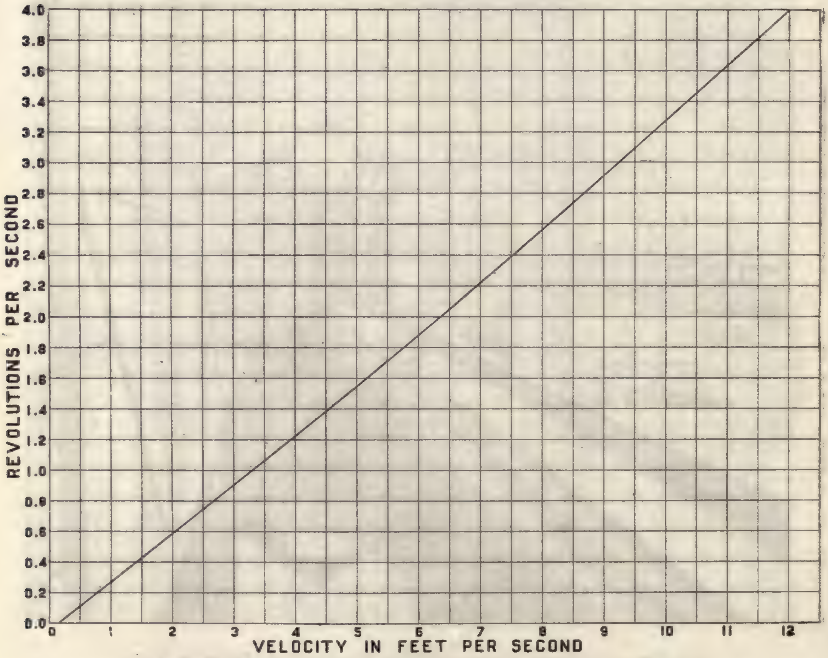

Fig. 70.—Current Meter Rating Curve (see page 121).

actual velocity of each of said parts. As a basis for the work, the cross-section of the stream should first be obtained by sounding. The vertical sections, chosen for the purpose of water observation, are usually five feet or more apart but the horizontal divisions are usually somewhat less as the varitions in the vertical velocities are usually much greater than in horizontal velocities. The size of both horizontal and vertical division depends on the irregularity of the distribution of velocity in the cross-section as well as on the accuracy required in the determination of flow. The greater the care used in the determination of the velocities in the unit areas and the

greater the number of such sub-divisions of the cross-section, the more accurate will be the work.

The observations of the velocity of flow by means of the current meter may be made in one of four ways:

First: By determining the velocity at frequent definite intervals of depth, and then ascertaining the amount of the average velocity in

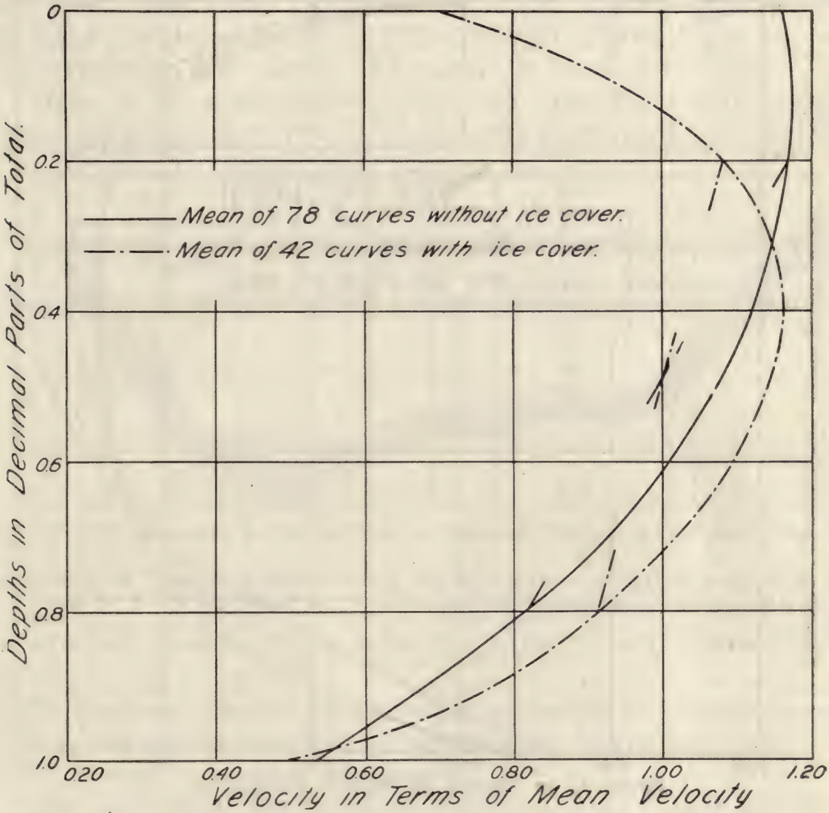

Fig. 71.—Comparison of Vertical Velocity Curves With and Without Ice Cover (see page 124).

each vertical section. This is known as the vertical velocity curve method.

Second: By determining the velocity at two points whose average is the average velocity in the vertical section, and consists of making determinations of the velocity at two-tenths and eight-tenths of the total depth respectively. This method has been compared with

the vertical velocity curve method and the average of the two determinations is found to give results very nearly the actual mean velocity both for open water and for ice conditions, as shown by Figs. 71, page 123 and 72.* This is known as the two point method.

Third: By making an observation of the velocity at a single point on the vertical corresponding to the depth of the thread of mean

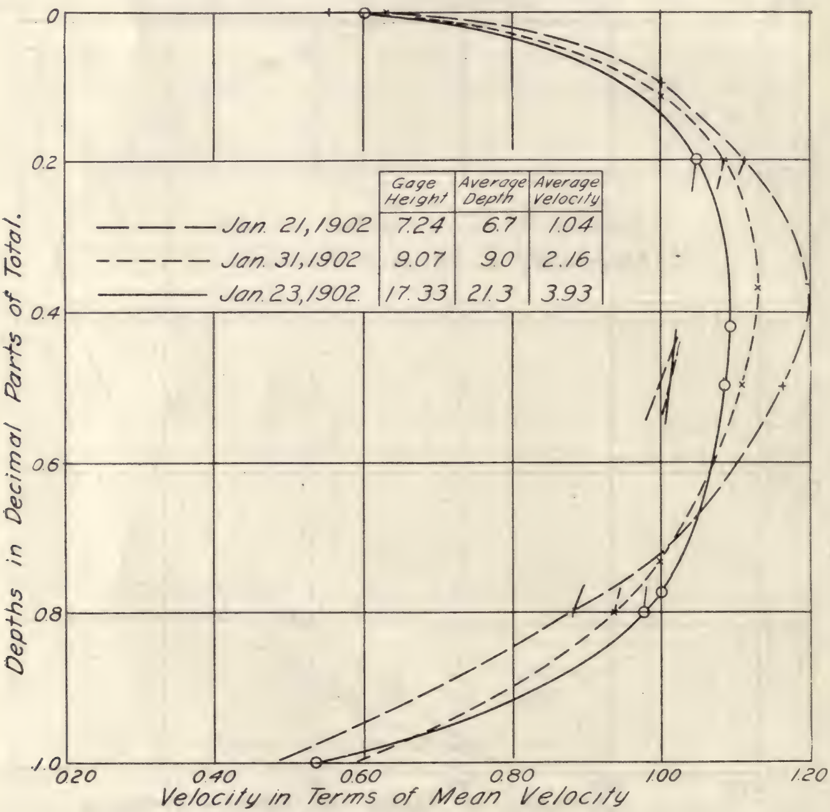

Fig. 72.—Vertical Velocity Curves Under Ice Cover, Wallkill River at New Paltz, N. Y.

velocity as indicated by investigations of the vertical velocity curves as determined for the section. Observations of velocity at some other depth than that at which the mean velocity occurs are sometimes made, the mean velocity to be used is deduced from the vertical

* See Water Supply and Irrigation Papers (U. S. G. S.) Nos. 187 and 337.

velocity curve applying to the same point and which was taken under similar conditions as exist during the single point observation. This is known as the single point method.

Fourth: By what is known as the integration method, which consists in lowering and raising the meter at a uniform speed from the surface to the bottom of the vertical and back, and noting the velocity indicated by the meter during the operation. The accuracy of the results obtained by this method have been questioned and, since it gives no data for the study of the vertical velocity curve and is inconvenient to apply during cold weather, it is not widely used.

Figure 73 shows the cross-section of the Saline River near Salina, Kansas, on September 30th, 1903, while the discharge measurements

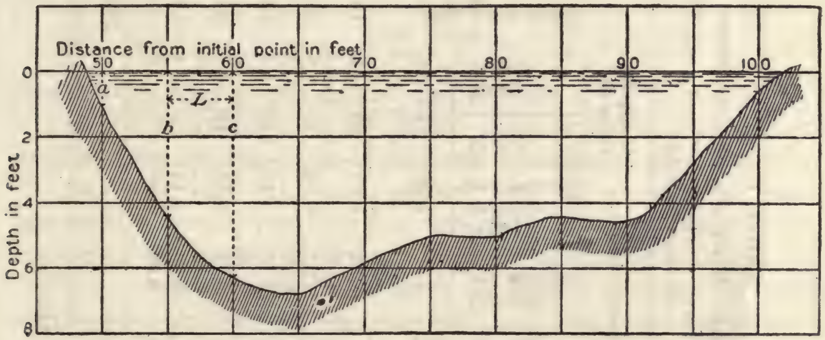

Fig. 73.—Cross-section of Saline River at Gauging Station Near Salina, Kan.*

recorded in Table 11 were being made. The soundings were taken at each five feet of width from the initial point and the velocity was observed at six-tenths depths below the surface in each of these verticals.

The discharge through each five-foot strip might be computed separately, but the computations are shortened by finding the discharge through each double strip at a time.

The mean depth and the mean velocity for the double strip of width ten feet are found from the formula:

$$d''_m = \frac{d_1 + 4d_2 + d_3}{6} \tag{55}$$

$$v''_m = \frac{v_1 + 4v_2 + v_3}{6} \tag{56}$$

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* From Water Supply and Irrigation Paper No. 94,—Hydrographic Manual by E. C. Murphy, J. C. Hoyt and G. B. Hollister. See page 47 et seq.

TABLE 11.

Gauging Made September 30, 1903, by E. C. Murphy. Meter No. 388, on Saline River Near Salina, State of Kansas.

[Gauge height: Beginning 7.96 ft., ending 7.96 ft. River stationary. Total area, 233 sq. ft. Mean velocity, 0.82. Discharge, 190 second-feet.]

Dist. from initial point.	Observations.				Velocity computations.				Computations of—		Area of sec.	Discharge of section.	Remarks. (On condition of channel, wind, equipment, gauge, boat, cable, methods, accuracy. Use cross-section pages in back of book for sketches.)
	Depth.	Depth of observation	Time in seconds.	Revolutions.	Total number of revolutions.	Revolutions per second.	Velocity per second.	Mean velocity per second.	Width.	Mean depth.			
49	0.9	....	....	....	......	....	0.0	0.20	1	0.6	0.6	0.1	Length of gauge wire measured and found to be 35.47 feet.
50	1.1	0.7	50	7 and 8	15	....	0.40	....	....	....	.....	......	Clear. No wind.
55	4.5	2.7	50	16 and 17	33	....	0.82	0.79	10	4.2	42.0	33.2	
60	6.3	3.8	50	21 and 22	43	....	1.06	....	....	....	.....	......	
65	6.8	4.1	50	19 and 18	37	....	0.92	0.95	10	6.6	66.0	62.7	
70	5.9	3.5	50	19 and 20	39	....	0.97	....	....	....	.....	......	
75	5.0	3.0	50	23 and 24	47	....	1.15	1.11	10	5.2	52.0	57.7	
80	5.0	3.0	50	22 and 23	45	....	1.10	....	....	....	.....	......	
85	4.4	2.6	50	14 and 15	29	....	0.73	0.73	10	4.5	45.0	32.9	
90	4.5	2.7	50	6 and 6	12	....	0.33	....	....	....	.....	......	
95	2.8	1.7	50	0 and 0	0	....	0.00	0.17	5	3.6	18.0	3.1	
100	0.6	0.4	50	0 and 0	0	....	0.00	0.00	7	1.3	9.1	0.0	
102	0.0	....	....	....	......	....	......	....	....	....	.....	......	
......	....	....	....	....	......	....	......	....	..	....	232.7	189.7	

Computed by E. C. Murphy. Checked by E. C. Murphy.

The discharge through the double strip is

$$q'' = d''_m\, v''_m\, 2L = \left(\frac{d_1 + 4d_2 + d_3}{6} 2L\right)\left(\frac{v_1 + 4v_2 + v_3}{6}\right) \tag{57}$$

Formulas (55) and (56) are based on the assumption that the stream bed is a series of parabolic arcs, also that the horizontal velocity curves are parabolic arcs, both of which assumptions are approximately true at good current-meter stations.

In computing the discharge and the mean depth through a single strip near the stream bank or a pier the mean velocity is found from the formulas:

$$v'_m = \frac{v_0 + v_1}{2} \tag{58}$$

$$d'_m = \frac{d_0 + d_1}{2} \tag{59}$$

where either v_0 or v_1 and d_0 or d_1 may be zero.

Velocity is computed to two places of decimals, mean depth, area, and discharge to one place of decimals for streams of ordinary size; for small streams with hard, smooth bottom, where depth can be measured to hundredths of a foot, the mean depth and area should be computed to two places of decimals and the discharge to one place.

These observations can be taken in shallow streams by wading or from a cable car (see Fig. 74, page 129), boat or bridge as the circumstances and conditions permit. A rope or cable, marked into suitable divisions and stretched across the stream, offers the best means of locating the horizontal points at which observations in the vertical planes are to be made.

Table 12 gives the results of a flow measurement of the Niangua River, Missouri, about forty miles south of its mouth. The measurement was made by wading, using a Price current meter held on a staff. Observations of velocity were made at two-tenths, six-tenths and eight-tenths of the depth and computations were made by the "single point" and "two point" methods previously explained. The computations of mean velocity were made by taking the simple arithmetical averages of the various points as shown in the table. The results of the two methods correspond within three second feet or a difference between results by the two methods of one and two-tenths per cent.

It is not believed that the mean velocity in a vertical is obtained with sufficient accuracy except by the most elaborate series of ob-

TABLE 12.

Gauging of the Niangua River, Mo., at Corkery Ford, June 18, 1914. Gauging Made by Wading with Price Current Meter.

Two Point Method.											Single Point Method.			
Distance from Initial Point.	Depth.	Ave. Depth.	Length of Section.	Area of Section.	Meter Readings. Revolutions per Second.			Velocity of Flow (from rating table) feet per Second.		Discharge q=av.	Meter Rev. per Sec. 6/10 Depth	Velocity of Flow, Feet per Sec.		Discharge q=av.
					2/10 Depth.	8/10 Depth.	Average.	In Vertical	Average for Section.			At Point.	Average.	
Feet	Feet.	Feet.	Feet.	Square Feet.						Cu. Ft. per sec.				Cu. Ft. per Sec.
0	0				0	0		0			0	0		
10	1.25	0.62	10	6.20	.528	.400	.464	1.05	.53	3.28	.468	1.06	.53	3.28
20	1.70	1.48	10	14.80	.635	.500	.568	1.28	1.17	17.32	.537	1.22	1.14	16.87
30	2.10	1.90	10	19.00	.793	.611	.702	1.57	1.43	27.18	.690	1.55	1.39	26.40
40	2.20	2.15	10	21.50	.848	.625	.737	1.65	1.61	34.62	.790	1.77	1.66	35.69
50	2.25	2.22	10	22.20	.862	.675	.769	1.72	1.68	37.30	.778	1.74	1.76	39.05
60	2.00	2.12	10	21.20	.870	.669	.770	1.73	1.73	36.68	.810	1.82	1.78	37.72
70	1.40	1.70	10	17.00	.810	.606	.708	1.58	1.65	28.04	.714	1.60	1.71	29.06
80	1.00	1.20	10	12.00	.737	.577	.657	1.48	1.53	18.36	.612	1.38	1.49	17.88
90	0.75	0.87	10	8.70	.618	.435	.526	1.18	1.34	11.65	.534	1.21	1.30	11.30
100	1.10	0.92	10	9.20	.430	.284	.357.	0.81	1.00	9.20	.361	0.82	1.02	9.39
110	1.20	1.15	10	11.50	.516	.404	.460	1.04	.93	10.70	.459	1.04	.93	10.70
115	1.10	1.15	5	5.75	.506	.400	.453	1.03	1.03	5.93	.465	1.05	1.04	5.98
120	1.10	1.10	5	5.75	.267	.146	.207	0.48	.76	4.37	.198	0.47	.76	4.37
125	1.00	1.05	5	5.25	0	0	0	0	.24	1.26	0	0	.24	1.26
	Total									245.9	Total			248.9

servations to justify any great mathematical refinement in computation. The latter method of computation is represented by the following:

$$(60)\quad q' = \text{discharge through a single section} = \left(\frac{d_1 + d_2}{2} L\right) \times \left(\frac{v_1 + v_2}{2}\right)$$

and the total discharge

$$(61)\qquad q = \Sigma q'$$

This method assumes that both the depth and velocity vary on straight lines, and affords a convenient means of calculation, particularly when the lengths of section are not constant, and gives good results although it is not based on theoretical considerations.

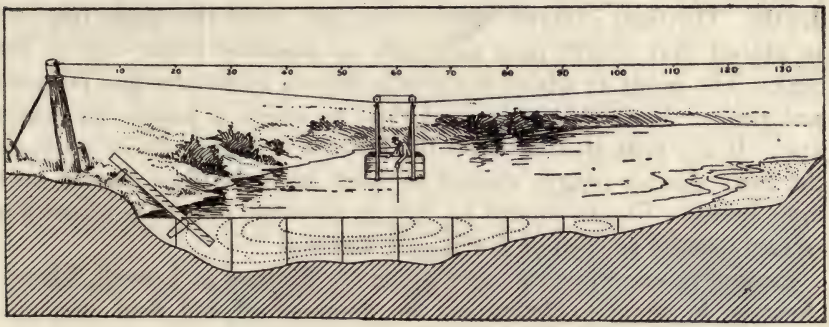

Fig. 74.—Cable Station, Car Guage, etc. (see page 127).

In using the vertical velocity curve to obtain the mean velocity in the vertical, the desired mean is obtained by dividing the area enclosed by the vertical curve and the vertical line representing zero velocity by the depth. For precise work the lines of equal velocity are drawn on the platted cross-section of the stream (see Fig. 54, page 96) and the separate areas enclosed by the curves are multiplied by the velocity which obtains across the area.

65. **Float Measurements.**—Where a single or only an occasional measurement of the flow of a stream is to be made, the use of floats is believed to be preferable, as under such conditions the calibration of the current meter and the exercise of necessary skill in its use are not apt to receive proper attention. Under such circumstances, therefore, float measurements are believed to be more accurate.

In the use of floats the writer usually selects round soft wood one to two inches in diameter and in various lengths, varying by about six inches. These are weighted at the lower end, usually by attaching pieces of lead pipe so that they will float with only about one to three inches of the rod exposed. To the exposed end is usually at-

tached small red or white streamers so that they may be readily seen and yet not be seriously affected by wind.

A point for the gauging is selected where the stream is fairly straight and uniform in section, and ropes, wires, or cables are stretched tightly across the stream, parallel to each other and twenty-five, fifty or 100 feet apart, as the location and velocity of the stream seem to demand. The ropes or wires should be tagged at intervals of five, ten or twenty-five feet, as the conditions seem to warrant, beginning at zero on the straight bank.

In starting the work a float is selected that will reach as near the bottom as possible without touching and should be about nine-tenths depth. The float is started five to ten feet above the upper line and so placed that it will pass as nearly as possible under one of the tags. The point at which it actually passes under the line is noted and recorded, also the point and time at which it passes the lower line. If the float should touch the bottom or a snag in its passage, the next shorter length should be used until the float passes both lines freely. Floats should be run at frequent intervals across the stream usually at each of the tagged stations.

Extensive experiments were made by Francis at Lowell, Massachusetts, in 1852, to determine the accuracy of rod float measurements.*

He found that discharge measurements based on the determination of velocities by floats were nearly always large as compared with measurements by a standard weir. This was due to the fact that the rod, on account of not reaching the bottom, was not affected by the low velocity near the stream bed and hence indicated too great a velocity. He found that the effect could be corrected by multiplying the discharge as obtained by the floats by a coefficient as follows:

(62) $q = Cq_1$ in which

q = actual discharge

q_1 = discharge as determined by floats.

(63) $C = \text{coefficient} = 1 - 0.116\,(\sqrt{D} - 0.1)$ and

(64) $D = \text{ratio}\ \dfrac{\text{distance of bottom of float from bottom of stream}}{\text{depth of stream.}}$

It will be observed that this coefficient C is always less than unity except where D is less than 0.01 which condition could not be possible in any natural stream.

* See "Lowell Hydraulic Experiments" by James B. Francis, pp. 146–208.

The Francis experiments were made in a channel of rectangular cross-section and floats of uniform length were used. In a natural stream the depth will vary at different points in the cross-section and floats of different lengths must be used. In such cases D will vary widely for the various floats used and to apply the correction, the velocity as determined by each float should be reduced by its particular constant C.

Experiments made at the Cornell hydraulic laboratory in 1900 by Kuichling, Williams, Murphy and Boright confirmed Francis' conclusion that rod float measurements are too large, only two out of thirty being smaller than measurements made by a standard weir. No attempt was made, however, to verify Francis' formula for the correction of such observations.†

In calculating the discharge from these measurements the average cross-section, in square feet, of each division is calculated and multiplied by the average velocity for the same in feet per second and the product will represent the discharge in cubic feet per second of the section represented by that float and the sum of the sections of all the floats will give the total discharge of the stream.

It is frequently desirable to calculate the discharge graphically, which may be done as shown by Fig. 75, page 132. This is done by plotting the two sections at the tag lines over each other and drawing in an average section between them. It is frequently desirable to draw in the floats in their true length and average position so that it may be seen at once how well the section was covered by the floats.

Under each float is laid off the velocity as determined by the same, to a selected scale, and a mean velocity curve is drawn through these points. By multiplying the ordinate of the velocity curve by the ordinates of the mean section, a quantity is obtained on the discharge curve which, when fully constructed, gives a discharge polygon, the area of which represents at the correct scale the discharge in cubic feet per second of the stream.

66. The Application of Stream Gaugings.—A single measurement of stream flow is of comparatively little value as a basis for estimating the continuous character of the flow of the stream, as will be seen by examination of any of the hydrographs previously shown. The flow of a stream, while it may appear to the casual observer

† See Water Supply and Irrigation Paper No. 95,—Accuracy of Stream Measurements, p. 54.

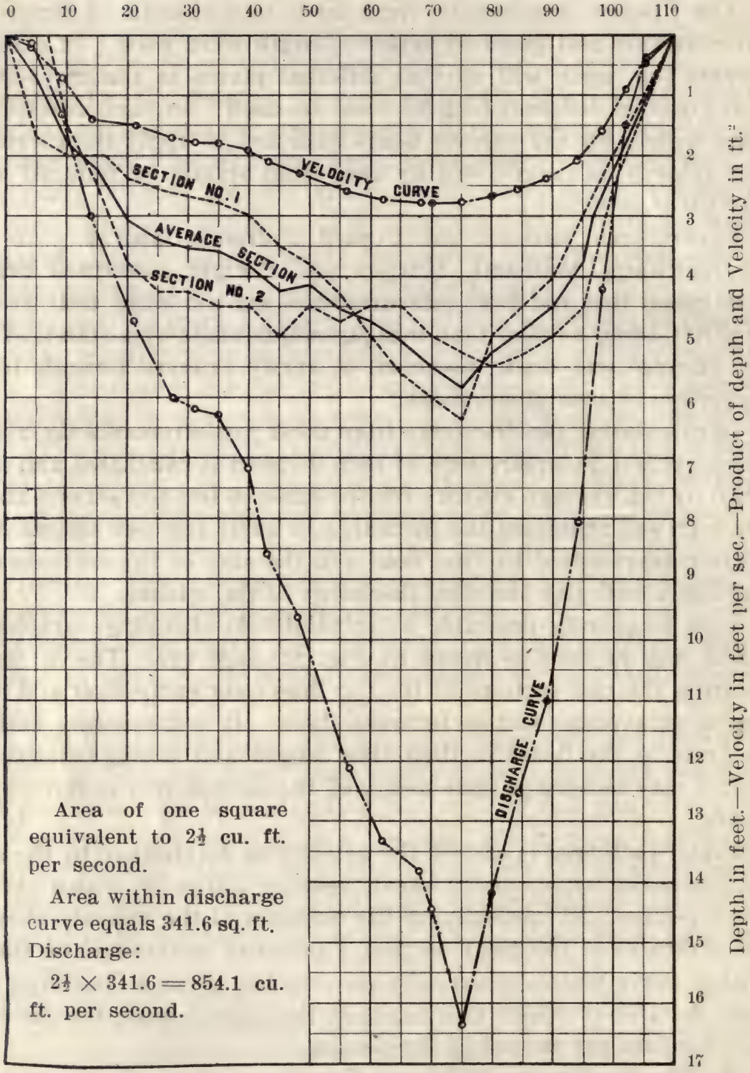

Fig. 75.—Graphic Determination of Stream Flow From Measurements (see page 131).

uniform, is actually subject to many and violent fluctuations and the flow may vary several hundred per cent. from minimum to maximum within a few days.

It has already been pointed out that in order to study the flow of a stream intelligently it is necessary to know the variations in flow

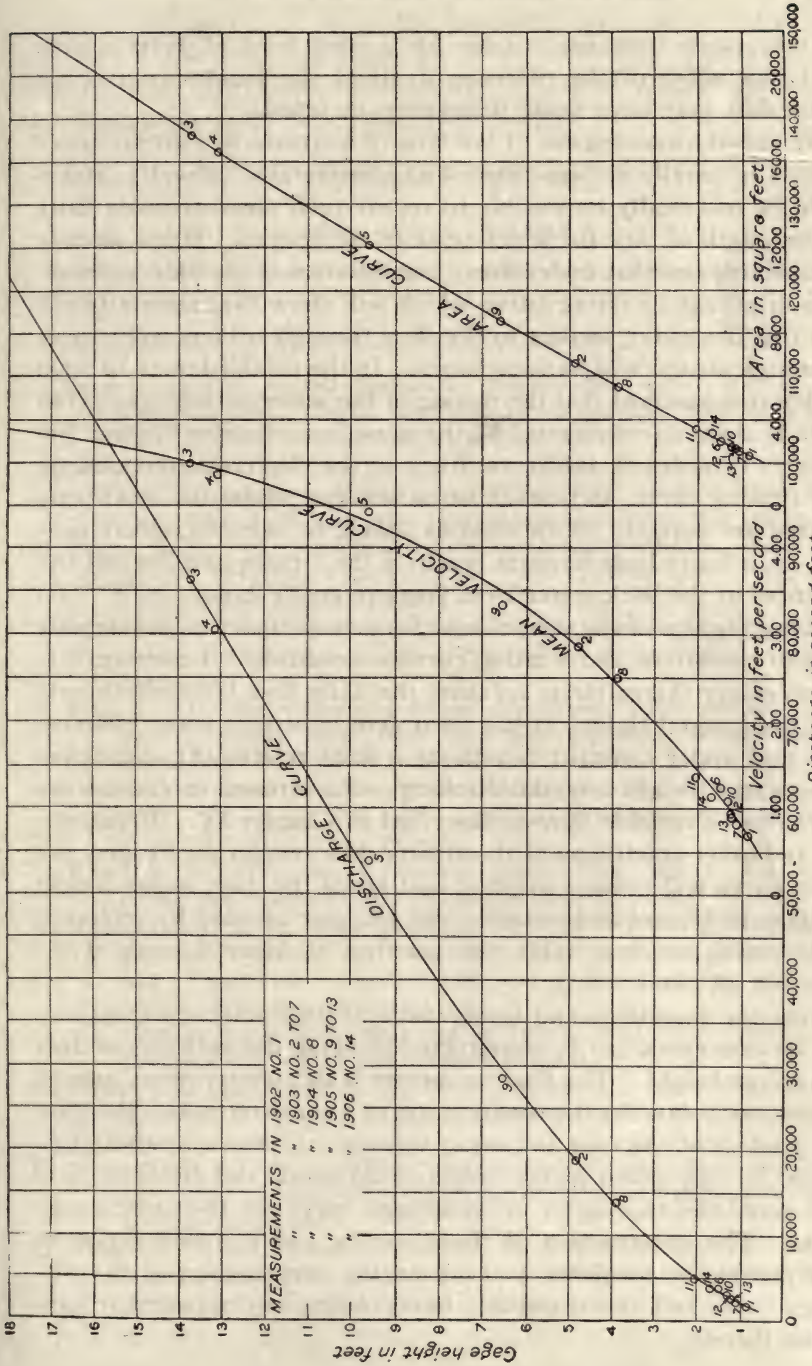

Fig. 76.—Discharge, Velocity and Area Curves for the Potomac River at Point of Rocks, Md. (U. S. G. S.). (See page 135.)

that take place from day to day for a long term of years during which the effect of the extreme of all of the factors controlling stream flow may have made themselves manifest.

The actual measurement of the flow of a stream by current meter or floats is usually accomplished with considerable difficulty, and it would be practically impossible to repeat such measurements daily for the length of time for which records are desired. It has already been pointed out that under many conditions it is possible to establish a discharge or rating curve which will show the relation of the height of the water surface to the flow through orifices over weirs or through channels of various forms. In the establishment of such relation it is assumed that the raising of the water surface to a given height is always accompanied by the same flow of water through the section. In order to assure accuracy in the observations based on such a rating curve, sections must be selected where the conditions assumed are correct. Such stations should be selected, where possible, on a fairly long uniform reach of the stream and beyond the influences of the back water from large rivers or dams.

After gaugings of the stream have been made under a considerable range of conditions and a rating curve is established therefrom, it is not necessary thereafter to measure the daily flow but only to note daily the gauge height. It has been determined by many observations that under constant conditions a fixed relationship exists between gauge height and the discharge of a stream, subject to the errors due to variable flow as described in Chapter IV. If the section and other conditions of the stream flow remain unchanged, the rating curve will remain constant and hence the daily gauge height can be quickly read and recorded and will give at once, by reference to the rating curve or table, the quantity of water flowing in the stream at all times.

From the soundings and levels made to determine the cross-section, an area curve can be constructed showing the variation of area with gauge height. The float or current meter observations furnish the necessary data for the construction of a curve of mean velocities. The product of the area and mean velocity, as shown by these two curves, for any given gauge height, must equal the discharge and must equal the reading of the discharge curve for the same gauge height. The construction of these curves, and a consideration of their properties, furnishes a check on the construction of the discharge curve and aids materially in correcting any apparent irregularities therein.

Figure 76, page 133, shows the discharge, mean velocity and area curves for the Potomac River at Point of Rocks, Maryland, and Fig. 64, page 102, shows the discharge and mean velocity curves for the Wallkill River at New Paltz, New York.

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CHAPTER VI

THE STUDY OF THE POWER OF A STREAM AS AFFECTED BY FLOW

67. Source of Water Power.—The amount of water power which may be developed from a stream in any particular case depends primarily on the flow of the stream and on the head that can be developed, maintained and utilized at the site proposed for the power plant. Both head and flow are essential for the development of water power, and both are variable quantities which are seldom constant for two consecutive days at any point in any stream. The variations in head and flow radically affect the power that can be generated by a plant installed for power purposes. These variations also greatly affect the power that can be economically developed from a stream at any locality. The accurate determination of both head and flow therefore becomes very important in considering water power installations and hence should receive the careful consideration of the engineer. The neglect of a proper consideration of either or both of these factors has frequently been fatal to the most complete success of water power projects.

68. Factors of Stream Flow.—The quantity of water flowing in a stream at any time, which is more briefly termed "stream flow" or "run-off," depends primarily upon the rainfall. It is, however, influenced by many other elements and conditions. It depends not only upon the total quantity of the yearly rainfall on the drainage area, but also on the intensity and distribution of the rainfall throughout the year. In addition to these factors the geological structure of the drainage area, the topographical features, the surface area of the catchment basin, the temperature, the barometric condition, all influence and modify the run-off. Sufficient data is not available for a full understanding of this subject, but enough is available so that the general principles involved can be intelligently discussed and the problems considered in such a way as to give a fairly satisfactory basis for practical work. A knowledge of the importance of the factors above mentioned and the extent to which they modify, influence or control stream flow, is essential to a broad knowledge of water power engineering. Space will not permit an extensive discussion of these factors in this volume but they have been more fully discussed by the auther in another volume (see Principles of Hydrology).

69. Broad Knowledge of Stream Flow Necessary.—The flow of a stream is constantly changing and any single measurement of that flow will not furnish sufficient data on which to base an intelligent estimate of the extent of its possible or even probable economical power development. A knowledge of the economical possibilities of such development must be based upon a much broader knowledge of the variations that take place in the flow of the stream. In order to fully appreciate the power value of a stream, the character and extent of its daily fluctuations must be known or estimated. Averages for the year, monthly averages, and estimates of average flow have often been taken as a basis for water power estimates, but they are more or less misleading, unsatisfactory and uncertain for the reason that such averages include extremes, the maximum of which are seldom available for water power purposes without more extensive storage than is usually practicable. These maximum and minimum flows which affect the power of a stream not only through the quantity flowing but also through their effect on head as well, as will be hereafter discussed, are of the utmost importance for a thorough consideration of water power. So also is a knowledge of the various stages of flow and the length of time that each will prevail. Such knowledge demands daily observations or estimates of daily flow for a considerable period.

70. Sources and Reliability of Hydrological Data.—In the literature listed at the end of this chapter is given the publication of the United States Geological Survey, which contains most of the available stream flow data of the streams of the United States. Some of these data have also been published in various state reports. The United States Weather Bureau also published data covering the daily river stages of many of the leading streams of the United States. While these publications do not include rating curves from which their gauge height can be translated into stream flow, they nevertheless furnish information of value for comparative study. The engineering department of the United States army also records gauge height and to some extent stream flow data on various navigable rivers of the United States. In many cases these data are unpublished, but are available at the local office of the engineering department. The water power branch of the Department of Interior of Canada also publishes stream flow data, covering the flow of various Canadian streams.

While the data from these various sources may be regarded as reasonably accurate, before they are used as a basis for proposed development, in which considerable investments are involved, they should be checked up as far as practicable.

Occasionally a dam, in which there is a considerable leakage, is used for measuring stream flow, and such leakage is ignored, although it seriously affects the apparent low water flow of the stream. Sometimes the coefficient used for calculating the discharge from a dam or river is not suitable for the type of over fall and the results are seriously in error. In other cases the rating section has been changed by filling or by excavation, until the rating curve especially for the state of low water is unsuited to the existing conditions. In still other cases serious errors have been made in establishing the rating curve or in setting the gauge from which the water height is determined or the gauge has been displaced and wrongly reset, so that the gaugings are erroneous.

The more important the project, the greater the care that should be taken to ascertain all available hydrographic data, and to so check the data available that their true value may be determined.

71. The Hydrograph and Its Uses.—On account of the great variation in stream flow, the data is often best studied in graphical form. The hydrograph, constructed for the study of stream flow and its influence on water power, may be drawn by representing the daily flow in cubic feet per second at the point of observation by the ordinates of the diagram and the element of time by the abscissas. The data may be platted chronologically as the flow actually occurs, or in the order of their magnitudes, showing the duration of the occurrence of each stage for the current year (see Fig. 77, page 142). The results are graphical diagrams which show the character and extent of the daily fluctuations and the proportional duration of stage in the flow of the stream at the point of observation during the period for which the hydrograph has been prepared.

In this hydrograph the irregular full line shows the discharge of the stream at Necedah, Wisconsin, based on the established rating curve. During the ice season it is known that the discharge, as shown by the water levels and rating curve, is too large on account of the effect of the ice on the flow (see section 50, page 99). As no ratings under ice conditions had been made at this station, the discharge during the ice season (December 1st to March 25th) were reduced by one-third to allow for this reduction in flow. The estimated flow during this period is therefore shown by the dotted line during the ice period. The chronological hydrograph shows a maximum flow for this year of 33,350 second feet, or 5.75 second feet per square mile, and a minimum flow of 1,160 second feet or 0.2 second feet per square mile on December 3.

The duration curve shows that a flow of 1,160 second feet was maintained for 100 per cent. of the year, 3,100 second feet for seventy-five per cent. of the year, and 5,150 second feet for fifty per cent. of the year.

It should be noted that any single observation of the flow of a stream represents a totally inadequate and unsatisfactory criterion for water power consideration. By reference to Fig. 77 it will be seen that, if the discharge of the Wisconsin River at Necedah had been measured only on August 5, 1904, the conclusion would have been reached that

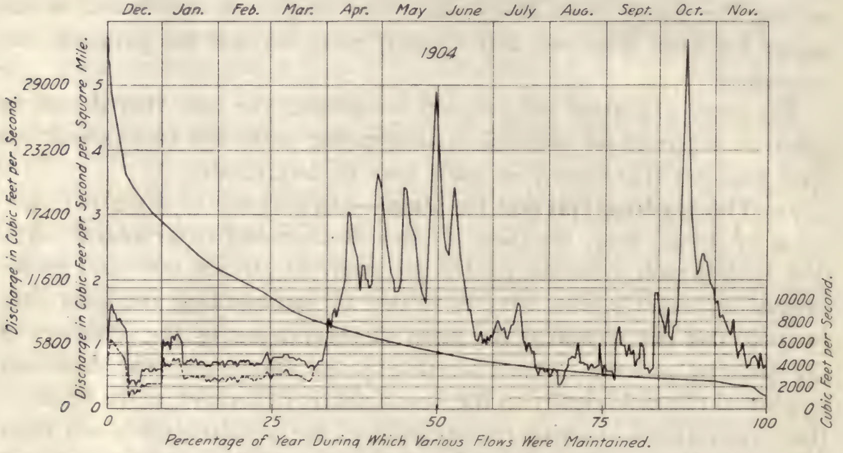


Fig. 77.—Discharge Hydrograph of the Wisconsin River at Necedah, Wis. (see page 141).

the discharge of the river was about 2,100 cubic feet per second. If the measurement had been taken only on August 15, 1904, the flow would have been determined at about 5,850 cubic feet per second, or almost three times as great as on the first date. The difference between any two dates might be even greater. *No single measurement nor any series of measurements for a single week or month would give a fair criterion from which the normal flow of a river can be judged.*

The hydrograph of the daily flow of a river for a single year gives a knowledge of the variation in flow for that year only under the peculiar conditions of the rainfall, the evaporation, and the other physical factors that modify the same and that obtain for that particular year. Such information, while important, is not sufficient for the purpose of a thorough understanding of the availability of the stream flow for power purposes. Observations show that stream flow varies greatly from year to year, and while, with a careful study of the influences of the various

factors on stream flow, together with a knowledge of the past variations in such factors, the hydrograph for a single year may give a fairly clear knowledge of the variations to be expected in other years where conditions differ considerably, still it is desirable that the observations be extended for as long a period as possible.

The relation of rainfall and other modifying factors, to stream flow are so indefinite that *only by a long series of actual observations, can the variations and the mean conditions be actually and accurately determined.*

72. **Comparative Hydrographs.**—Hydrographs based on observations made at other points on the same river, or on other adjacent rivers where conditions are fairly similar, are of great value in considering the local stream flow,—provided all modifying conditions are understood and carefully considered. Hydrographs are ordinarily prepared to show the actual flow at the point at which observations are made. If the observations (and the hydrographs based thereon) made at some other point on a stream, or on some other streams, are to be used for the consideration of the flow at a point where a water power plant is to be installed or considered, the relation of the flows at the several points must be determined.

As a basis for such comparison of stream flow, it may be assumed that the flow per unit of area at different points on the same stream, or at points on different streams under similar circumstances, is essentially the same or is in some definite proportion. This is not strictly true, or perhaps it may be more truly said that the apparent similarity of conditions is only approximate and hence errors in comparative results must necessarily follow. For a satisfactory consideration of the subject of comparative hydrographs the variations from these assumptions must be understood and appreciated.

A comparison of streams in different parts of the country widely separated are seldom warranted, and frequently the flow of streams in the same state is widely different. For example, the Wisconsin River has a low water, high water and mean flow per square mile about double that of the Rock River in the same state, and the flow of the Hudson River, the Oswego River and the Genesee River in the state of New York are widely different and not at all suitable for a basis of estimates of each others flow. When conditions are fairly comparative the assumption of similarity of flow is often essentially correct and in such cases comparative hydrographs will afford a basis for an intelligent consideration of stream flow where local hydrographs are not available. Fig. 77 is a hydrograph constructed from observations made on

Fig. 78.—Drainage Area of Wisconsin River Above Kilbourn, Wis.

the Wisconsin River at Necedah, Wisconsin, by the United States Geological Survey for the water year, 1904, and shows the daily rate of discharge of the Wisconsin River at that point for the year named. The area of the Wisconsin River (see Fig. 78, page 144) above Necedah is 5,800 square miles. If, therefore, a horizontal line be drawn from the point representing 5,800 cubic feet per second on the discharge scale (see Fig. 77), the line so drawn will represent a discharge at Necedah of one cubic foot per second per square mile of drainage area, and a similar line drawn from the 11,600 cubic foot point on the ver-

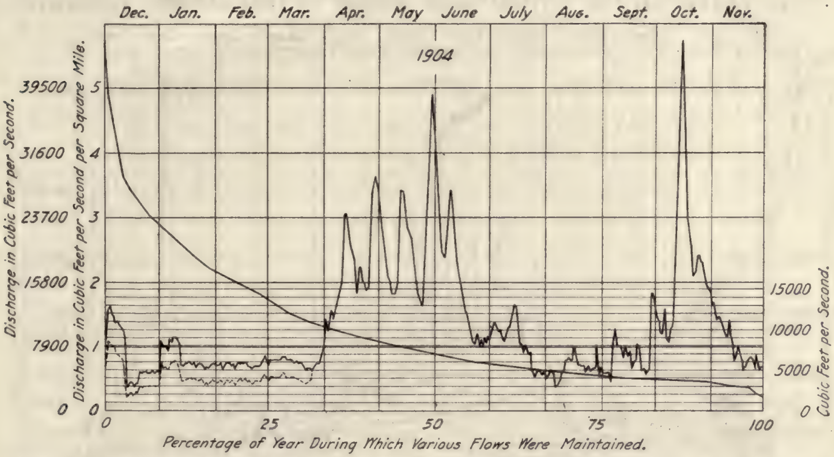


Fig. 79.—Hydrograph of Wisconsin River Based on Observations at Necedah, Wis., Adapted by Change of Scale to Conditions at Kilbourn, Wis. (see page 146).

tical scale will represent a discharge of two cubic feet per second per square mile, and so on. These lines may be fairly regarded not only as indicating the flow per unit of area of the river at Necedah, but also the relative flow per unit of area of the Wisconsin River at points not greatly distant therefrom. At Kilbourn (see Fig. 78), located on the same river about forty miles below Necedah, the flow may be assumed to be similar and proportionate to the flow at Necedah. Above Kilbourn the drainage area is 7,900 square miles, and with similar flow the discharge would be proportionately greater. The fact must be recognized, and acknowledged, that the hydrograph is strictly applicable only to the point at which it is taken, and that certain errors will arise in considering its application to other points, yet observations and comparisons show that, while such errors exist, they are not nearly so important as the errors which arise from the consideration of averages, either annually or monthly.

Consider, therefore, on this basis the Necedah hydrograph as shown in Fig. 77. On this diagram a flow of one cubic foot per second per square mile at Necedah, representing an actual flow of 5,800 cubic feet per second at that point, would, by proportion, represent a flow of 7,900 cubic feet per second at Kilbourn and, with a suitable change in scale, the diagram may be redrawn to represent the flow at Kilbourn as shown in Fig. 79, page 145. This same method can be applied to any point on the same river or to comparative points on different rivers.

73. The Hydrograph as a Power Curve.—The hydrograph, by a simple change in the vertical scale similar to that already considered,

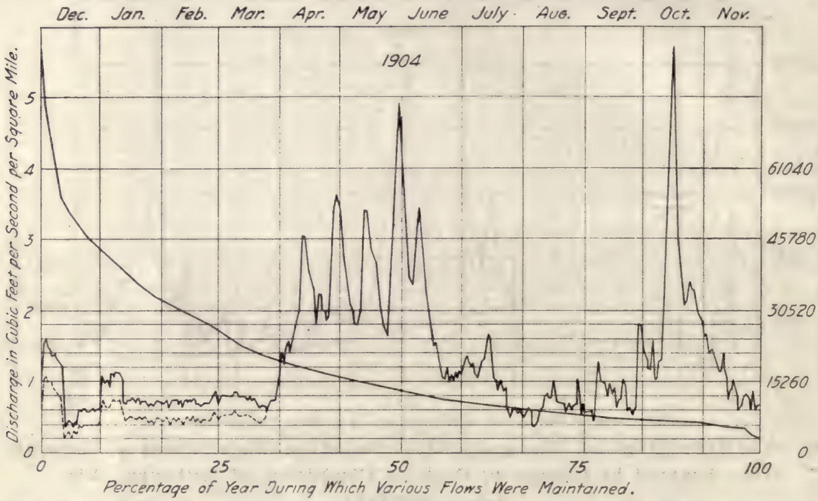


Fig. 80.—Power Hydrograph Showing Continuous (24 Hour) Theoretical Horse Power at Kilbourn, Wis. (17 Foot Head).

may also be made to show graphically the variations in the power of the stream. If, for example, at Kilbourn, a constant fall of seventeen feet be assumed, then a flow of one cubic foot per second per square mile represents a total flow of 7,900 cubic feet per second, and this flow, under seventeen foot head, will give a theoretical hydraulic horse power as follows:

$$\text{H. P.} = \frac{7900 \times 17}{8.8} = 15261$$

Now if a hydrograph be constructed on such a scale that the line of flow of one cubic foot per second per square mile will also represent 15,261 H. P., the result will be a power hydrograph (see Fig. 80), which represents the continuous (twenty-four hours per day) theoretical power of the river under the conditions named.

On account of losses in the development of power the full theoretical power of a stream cannot be developed, and hence the actual power that can be realized is always less than the theoretical power of the stream. If it is desired to consider the actual power of the stream on the basis of developing the same with turbines of eighty per cent. efficiency, the line representing the flow of one cubic foot per second per square mile will represent the actual horse power to an amount determined as follows:

$$\text{A. H. P.} = \frac{7900 \times 17 \times .80}{8.8} = \frac{7900 \times 17}{11} = 12209$$

A hydrograph platted so that the line of one cubic foot per second per square mile will represent this amount, will indicate the actual horse

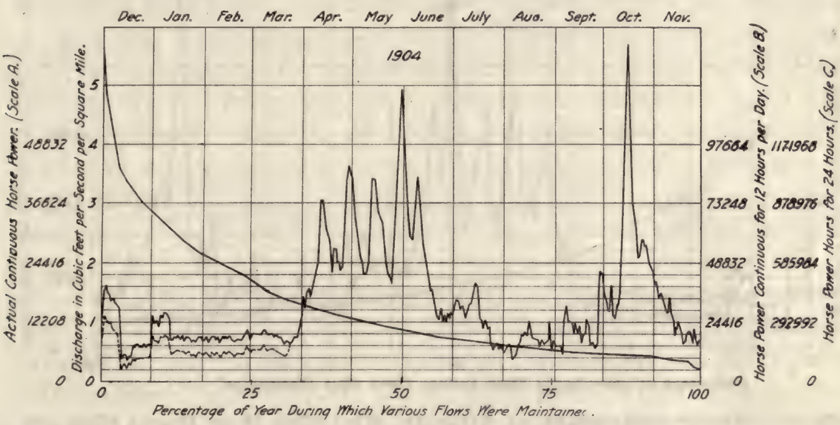


Fig. 81.—Power Hydrograph at Kilbourn, Wis. (17 Foot Constant Head).

power of the river at Kilbourn with the wheels working with the efficiency and under the head named. Such a hydrograph is shown by Fig. 81, referred to by the left-hand scale (A). Power, however, is not always used continuously for twenty-four hours. If pondage is available the night flow may be stored and utilized during the day. If the flow of twelve hours at night is impounded and used during the day under the seventeen foot head, the power will be double that shown on scale A, and can be represented by another change in scale as shown by Fig. 81, referred to scale B. If the flow for the fourteen hours of night is stored and utilized in the ten hours of day, then the hydrograph can be made by another change in scale to represent the ten hours power as shown by Fig. 82, page 148.

The total horse power hours which are available from a stream for each day may be represented (either theoretically or actually) by multiplying the scale of continuous power by twenty-four. The actual horse power available at Kilbourn under the conditions named is represented by scale C in Fig. 81. It will be noted that by pointing off one place in the figures of scale C, Fig. 81, the hydrograph will represent the same condition as shown in Fig. 82.

74. The Practical Study of Stream Flow.—When local observations are available for extended periods, such long time observations

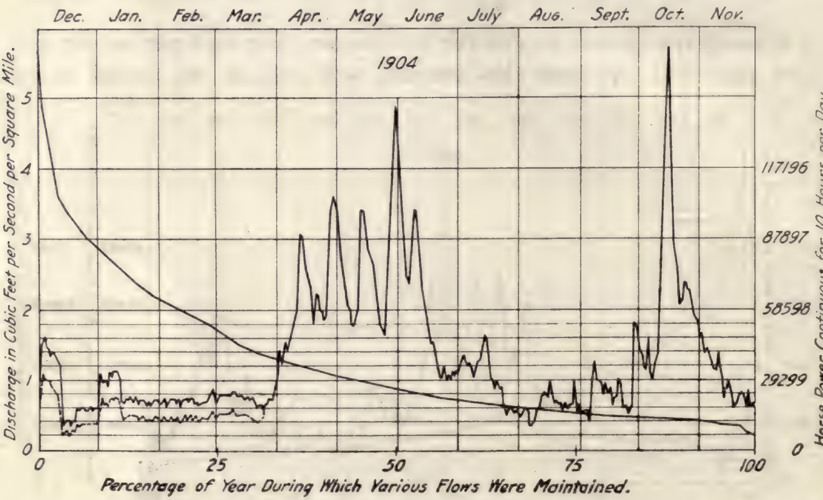


Fig. 82.—Power Hydrograph at Kilbourn, Wis. (Ten Hour Power With Perfect Pondage and 17 Foot Constant Head).

may remove the estimates of flow entirely from the domains of speculation and place them on the solid ground of observed facts. Hydrographs of a river that cover the full range of conditions of rainfall, temperature, etc., which are liable to prevail on its drainage area, give a very complete knowledge of the flow of the stream for the purpose of the consideration of water power.

It is rare, however, that observations of stream flow for a long term of years are available at the immediate site of a proposed power plant. Such observations are ordinarily made only at locations where power has been developed and where water power or similar interests have been centered for a long period of time. Occasionally, however, the future value of potential powers is recognized and appreciated, and local observations are maintained for a series of years by interested

parties, having a sufficient knowledge of the subject to recognize the value and importance of such information. The variation of flow for some considerable time previous to construction is thus available upon which to base the design.

In considering new installations, one of four conditions obtains:

First: Hydrographs are available at the immediate site proposed.

Second: Hydrographs are available at some other point on the river above or below the proposed installation.

Third: Hydrographs are not available on the river in question but are available on other rivers where essentially similar conditions of rainfall and stream flow prevail.

Fourth: No hydrographs, either on the river in question or on other rivers of a similar character and in the immediate vicinity, are available.

75. **Local Hydrographs.**—When hydrographs, constructed from observations taken at the immediate site of the proposed water power installation, are obtainable, for a considerable number of years, the most satisfactory character of information is available for the consideration of a water power project. Under such conditions the engineer is not obliged to consider the relation of rainfall to run-off or to speculate as to the relative value of the stream in question compared with other adjacent streams, or as to the effects of the physical conditions of drainage area, evaporation, temperature and other factors on stream flow. The actual flow of the stream from day to day, perhaps through all ranges of rainfall, temperature, evaporation and other physical conditions, is known and the principal points which must be considered are: *First,* the head available; *Second,* the effects of the variations of flow on the variations in head; and *Third,* the extent to which the flow can be economically developed or utilized. Generally, however, even where local hydrographs are available, they are not sufficiently extended to cover all the variations in river flow which must be anticipated, and it is ordinarily desirable to compare the available data with the flow at other points on the stream in question or with other streams in the immediate vicinity.

76. **The Study of Local Hydrographs.**—As an example of the consideration of flow data available at the immediate site of a proposed hydraulic development, let the possibilities of the development of the Fox River diversion at the Rapide Croche Dam be investigated. The drainage area of the Fox River above the outlet of Lake Winnebago (see Fig. 83, page 150) is about 6,200 square miles. As the lower Fox has few tributaries this average area may be regarded as common to any power location on the lower Fox River.

Fig. 83.—Drainage Area of the Fox River and Its Tributaries (see page 149).

The Green Bay and Mississippi canal, which connects Green Bay with Lake Winnebago in Wisconsin, parallels the Fox River and is owned and operated by the United States government. The Fox River has been gauged by the United States army engineers at the Rapide Croche Dam for many years. The gaugings prior to 1895 are questionable, as the dam was not maintained in good order, but since that date attention has been given to the maintenance of the dam, and the gaugings are believed to be fairly accurate. For the purpose of this study they may be assumed to be correct, although the matter should be investigated if the data is to be used as the basis of any considerable investments.

The gaugings show the amount of water flowing in the river, and available for power purposes from 1895 to 1913 inclusive. The amount discharged through the canal is not known, but it is not large, and will probably not materially increase. The annual second-feet hydrographs showing the rate of daily flow at the Rapide Croche Dam are shown in Figs. 84 and 85, pages 152 and 153.

From the daily records of flow the average shortage of water has been estimated for each one-tenth of a cubic foot per second per square mile from one-tenth to seven-tenths and for the entire length of time for which records are available. Tabulations of the results of the investigation for three-tenths and seven-tenths cubic feet per second per square mile are shown in Tables 13 and 14, pages 154 and 155.

In the last column of each table is shown the average deficiency in the same units for each year, and at the foot of this last column the average deficiency for the entire period of nineteen years.

Table 15, page 156, is a tabulation of the maximum and minimum flows of the Fox River. The greatest discharge recorded in the nineteen-year period occurred on April 16, 1906, and reached 15,919 second feet, or 2.547 cubic feet per second per square mile on a drainage area of 6,250 square miles. This maximum was closely approached on June 6 of 1895 and 1905 and again on April 11, 1913. The lowest maximum flood of any year occurred on June 11, 1896, with a discharge of 4,605 second feet, or .737 cubic feet per second per square mile.

On July 18 and again on August 18, 1910, and during a period extending from August 1, to August 10, 1911, no flow was recorded over the government dam at Rapide Croche. Some water was used for navigation purposes in the locks around the rapids, but as a supply for water power at this point, the flow of the river must be considered zero

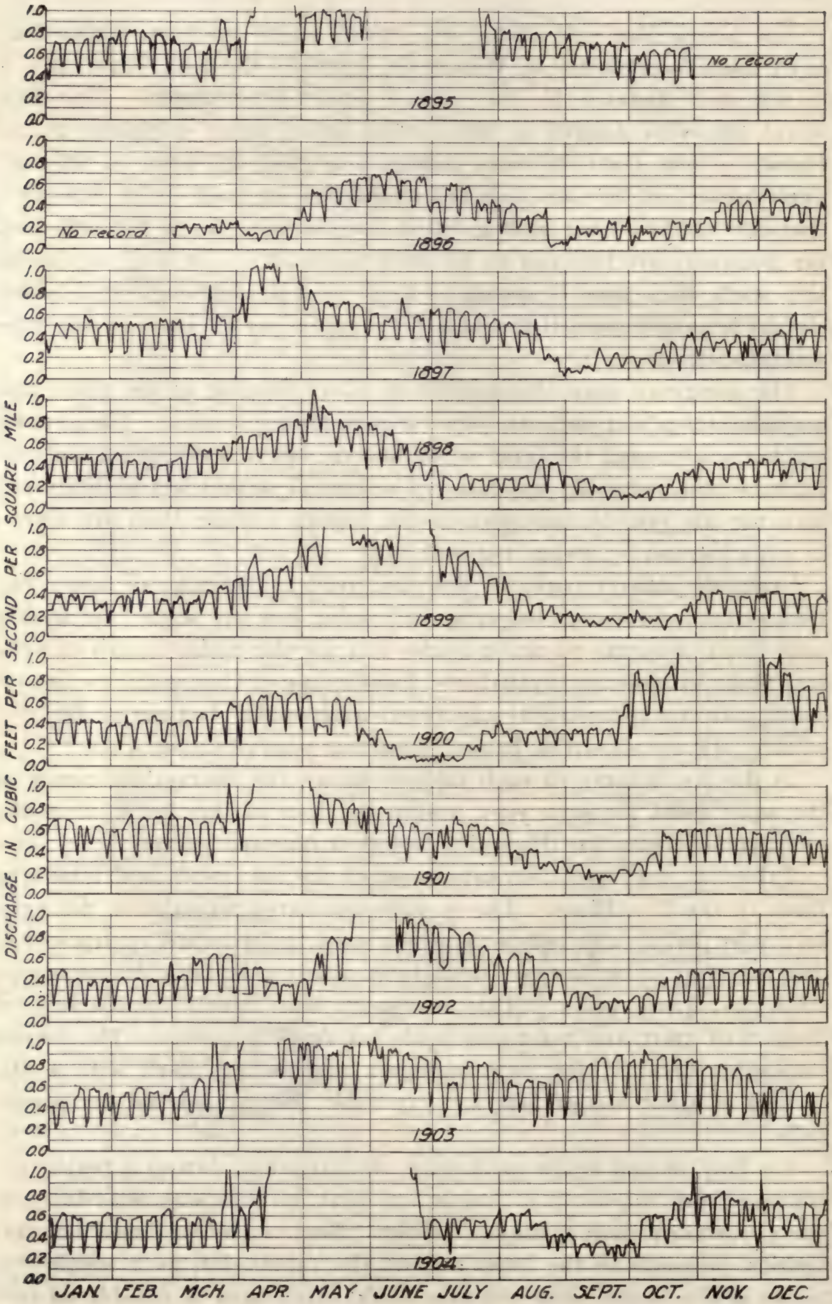


Fig. 84.—Discharge Hydrographs of the Fox River at Rapide Croche Dam for the Years 1895 to 1904, Inclusive (see page 151).

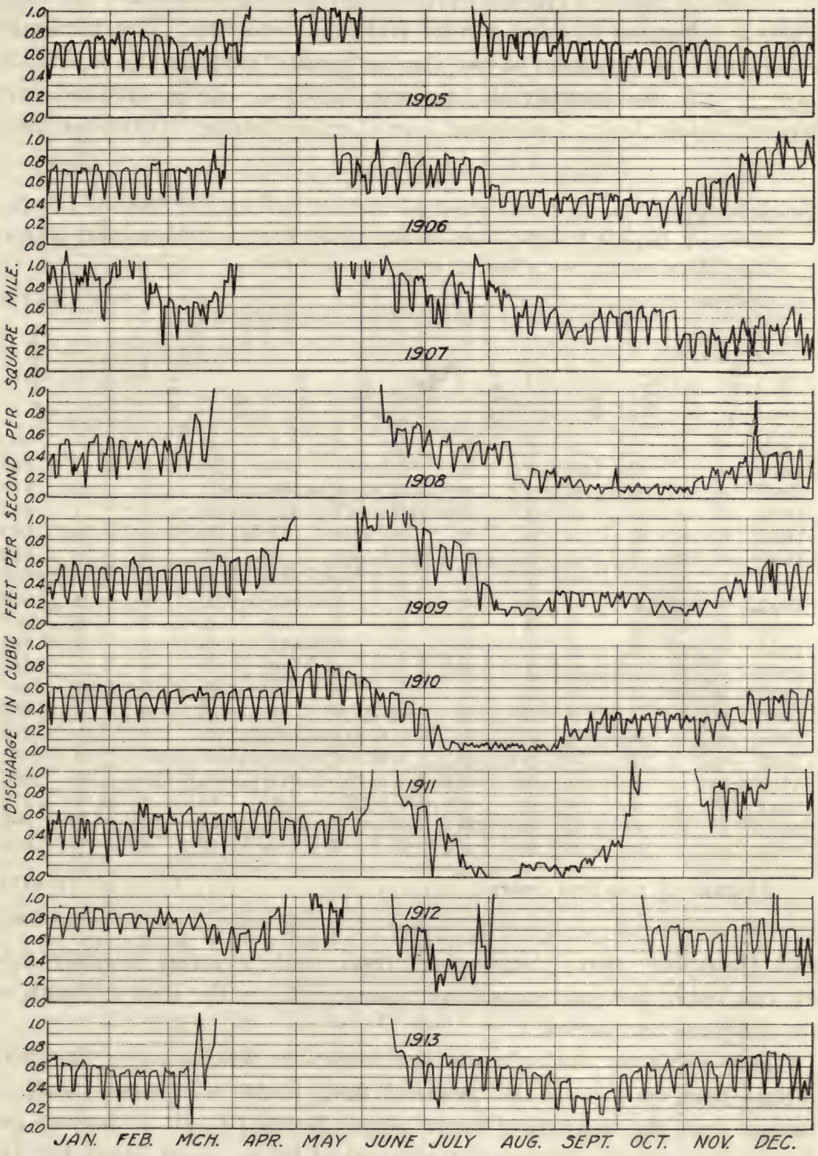


Fig. 85.—Discharge Hydrographs of the Fox River at Rapide Croche Dam for the Years 1905 to 1913, Inclusive (see page 151).

on the above dates. The next lowest flow was on September 28, 1896, when a minimum of nine second feet was recorded. The minimum flows, with the exception of one year in January and another in March, have been in the summer and autumn months. July is credited with three; August, four; September, four; October, three; November, two;

TABLE 13.

Deficiencies in Flow in Cubic Feet per Second per Square Mile of the Fox River at Rapide Croche Dam at 0.3 Cubic Feet per Second per Square Mile (Equivalent to a Flow of 1875 Second Feet).

Year.	Jan.	Feb.	Mar.	Apr.	May.	June.	July.	Aug.	Sept.	Oct.	Nov.	Dec.	Average.
1895	0	0	0	0	0	0	0	0	0	0	.....	.....	0
1896	...	...	0	0.148	0	0	0	0.063	0.276	0.128	0	0	0.0615
1897	0	0	0	0	0	0	0	0	0.166	0.070	0	0	.0197
1898	0	0	0	0	0	0	0.047	0.007	0.125	0.106	0	0	.0237
1899	0	0	0	0	0	0	0	0.004	0.141	0.115	0	0	.0217
1900	0	0	0	0	0	0.159	0.146	0.005	0	0	0	0	.0258
1901	0	0	0	0	0	0	0	0	0.103	0	0	0	.0086
1902	0	0	0	0	0	0	0	0	0.096	0.007	0	0	.0086
1903	0	0	0	0	0	0	0	0	0	0	0	0	.000
1904	0	0	0	0	0	0	0	0	0.001	0	0	0	.0001
1905	0	0	0	0	0	0	0	0	0	0	0	0	.000
1906	0	0	0	0	0	0	0	0	0	0	0	0	.000
1907	0	0	0	0	0	0	0	0	0	0	0	0	.000
1908	0	0	0	0	0	0	0	0.038	0.184	0.220	0	0.101	.0452
1909	0	0	0	0	0	0	0	0.156	0.058	0.105	0.059	0	.0315
1910	0	0	0	0	0	0	0.201	0.251	0.071	0.006	0.032	0	.0467
1911	0	0	0	0	0	0	0.056	0.232	0.121	0	0	0	.0381
1912	0	0	0	0	0	0	0	0	0	0	0	0	.000
1913	0	0	0	0	0	0	0	0	0	0	0	0	.000
Average of nineteen years													0.01743

and December, two. September, 1896, with a mean monthly flow of .024 cubic feet per second per square mile, is the lowest month of the nineteen-year period.

Table 16 shows not only the deficiencies in flow but also the theoretical horse power of the assumed flow, under a constant head of twenty feet, the theoretical horse power hours (T. H. P. hours) per annum, the annual deficiency in theoretical horse power hours per annum and the percentage of such deficiency to the total annual power. This table therefore shows the amount of auxiliary power that would be theoretically necessary to maintain a continuous output of from

TABLE 14.

Deficiencies in Flow in Cubic Feet per Second per Square Mile of the Fox River at Rapide Croche Dam at 0.7 Cubic Feet per Second per Square Mile (Equivalent to a Flow of 4375 Second Feet).

Year.	Jan.	Feb.	Mar.	April.	May.	June.	July.	Aug.	Sept.	Oct.	Nov.	Dec.	Average.
1895	0.066	0.003	0.063	0.000	0.000	0.000	.000	0.000	0.057	0.149	......	......	0.0338
1896	.....	....	.493	.548	.194	.099	.250	.463	.676	.528	0.376	0.318	.3945
1897	.255	.254	.263	.000	.052	.176	.184	.397	.566	.470	.400	.327	.2787
1898	.287	.320	.221	.042	.000	.181	.447	.407	.525	.506	.343	.349	.3023
1899	.393	.365	.337	.110	.000	.000	.089	.404	.531	.515	.358	.371	.2894
1900	.349	.338	.288	.149	.220	.559	.546	.405	.374	.000	.000	.000	.2690
1901	.131	.091	.081	.000	.000	.100	.135	.349	.503	.289	.175	.254	.1757
1902	.335	.355	.234	.323	.000	.000	.006	.233	.496	.407	.314	.333	.2530
1903	.255	.224	.083	.000	.000	.000	.035	.144	.003	.000	.105	.235	.0903
1904	.204	.195	.152	.000	.000	.000	.199	.219	.401	.142	.046	.116	.1395
1905	.067	.004	.053	.000	.000	.000	.000	.000	.057	.150	.112	.141	.0487
1906	.094	.069	.000	.000	.000	.007	.031	.264	.313	.344	.137	.000	.1049
1907	.000	.000	.068	.000	.000	.000	.000	.109	.287	.235	.387	.331	.1181
1908	.293	.259	.000	.000	.000	.000	.233	.438	.584	.620	.333	.501	.2717
1909	.274	.261	.228	.035	.000	.000	.109	.556	.458	.505	.459	.228	.2594
1910	.194	.194	.185	.172	.021	.232	.601	.651	.471	.406	.432	.270	.3191
1911	.246	.215	.196	.147	.219	.000	.456	.632	.521	.000	.000	.000	.2193
1912	.000	.000	.017	.000	.000	.000	.299	.000	.000	.000	.089	.077	.0402
1913	.146	.218	.000	.000	.000	.000	.121	.190	.382	.173	.138	.100	.1223
Average for nineteen years													0.1963

1,420 T. H. P. to 9,940 T. H. P. and the equivalent daily output in theoretical horse power hours, provided there is sufficient pondage to take care of the fluctuations in daily load.

As the hydrograph shows that at certain times no flow is available it is evident that in order to maintain power at all times an auxiliary steam plant would have to be installed with a capacity as great as that

TABLE 15.

Maximum and Minimum Discharges of the Fox River at Rapide Croche Dam for Each Year from 1895 to 1913 Inclusive.

Year.	Date.	Max. Flood.		Date.	Min. Discharge.	
		cu. ft. per sec. per sq. mi.	cu. ft. per sec.		cu. ft. per sec. per sq. mi.	cu. ft. per sec.
1895..	June 6......	2.467	15416	Mar. 14.........	0.330	2062
1896..	June 11.....	.737	4605	Sept. 28.........	.001	9
1897..	Apr. 23.....	1.397	8728	Aug. 30.........	.019	116
1898..	May 6......	1.096	6852	Oct. 9..........	.061	383
1899..	May 17.....	1.403	8767	Dec. 25..........	.017	105
1900..	Nov. 3......	1.536	9597	July 4..........	.021	131
1901..	Apr. 12.....	1.925	12033	Sept. 22.........	.180	675
1902..	May 29......	1.971	12317	Oct. 5..........	.070	435
1903..	Apr. 5.......	1.488	9297	Jan. 4...........	.193	1206
1904..	May 13......	1.869	11682	Sept. 22.........	.158	988
1905..	June 6......	2.467	15416	Dec. 25..........	.297	1851
1906..	Apr. 16.....	2.547	15919	Oct. 22..........	.130	811
1907..	Apr. 11.....	1.950	12190	Nov. 3..........	.122	766
1908..	May 21......	2.086	13037	Aug. 23.........	.031	194
1909..	May 8.......	2.092	13075	Nov. 7..........	.060	375
1910..	Apr. 27.....	.851	5319	July 18 & Aug. 18	.000	0
1911..	Dec. 19......	1.932	12075	Aug. 1 to Aug. 10	.000	0
1912..	May 30......	1.858	11612	July 4...........	.118	738
1913..	Apr. 11......	2.350	14874	Sept. 22.........	.015	97

of the hydraulic plant, unless the industry for which the plant is to be installed can utilize the power as available, closing down partially or wholly on occasion as the flow decreases.

The extent to which a development should be made will depend on the value of the power, the necessity and cost of auxiliary power, the cost of development and the nature of the industry for which the power is to be utilized.

On account of the load factor of every industry, constant power is never fully utilized and there are numerous irregularities in loads which

TABLE 16.

Discharge.		Average Deficiency.								
Cubic feet per second per square mile.	Cubic feet per second.	Cubic feet per second per square mile.	Cubic feet per second.	Millions of cubic feet per annum.	Average theoretical continuous horsepower at the assumed discharge under 20 ft. head.	Average deficiency in theoretical continuous horsepower at actual discharge under 20 ft. head.	Average theoretical H. P. hours per annum at the assumed discharge under 20 ft. head.	Average deficiency in theoretical H. P. hours per annum at actual discharge under 20 ft. head.	Percentage of deficiency.	Average theoretical H. P. hours per annum at actual discharge under 20 ft. head.
0.1	625	0.00102	6.4	201.8	1420	15	12,439,200	131,400	1.0+	12,307,800
0.2	1250	.00538	33.6	1,059.6	2840	76	24,878,400	665,760	2.6—	24,212,640
0.3	1875	.01743	108.9	3,434.3	4260	270	37,317,600	2,365,200	6.3+	34,952,400
0.4	2500	.04110	256.9	8,101.6	5680	584	49,756,800	5,115,840	10.3—	44,640,960
0.5	3125	.07699	481.2	15,175.1	7100	1094	62,196,000	9,583,440	15.4—	52,612,560
0.6	3750	.12954	809.6	25,531.5	8520	1840	74,635,200	16,118,400	21.6+	58,516,800
0.7	4375	.19631	1226.9	38,691.5	9940	2788	87,074,400	24,422,880	28.1—	62,451,520

must be considered as influencing output and possibly influencing somewhat the amount of auxiliary power which may be actually needed.

The influence of the variation in the demand for power may be considered as follows:

In an installation recently contemplated, the variation in load was estimated as follows:

Average daily load	100 per cent.
Average daily load maximum month	130 per cent.
Maximum daily load	150 per cent.
Peak load maximum day	200 per cent.
Average daily load minimum month	85 per cent.
Average load of minimum day	65 per cent.

Let this load be superimposed on the mean annual hydrograph (see Fig. 86, page 159). In Fig. 86 the irregular line *a a a* shows the variation of the mean annual hydrograph. The average power output for which the plant is to be designed is shown by the straight line *b b b,* and the irregular, average monthly load which must be anticipated (as assumed for the conditions existing in this case), is shown by the stepped line *c c c.*

The limit to the variations in the demand for power as estimated above is also shown by horizontal lines duly indicated. By various cross hatching the diagram indicates:

First: The average water available to maintain average power but unutilized.

Second: The water available to develop average power, but which will probably be wasted (without storage) because less than average power will be used at that period.

Third: The auxiliary power required to develop the average power during periods of deficient water supply.

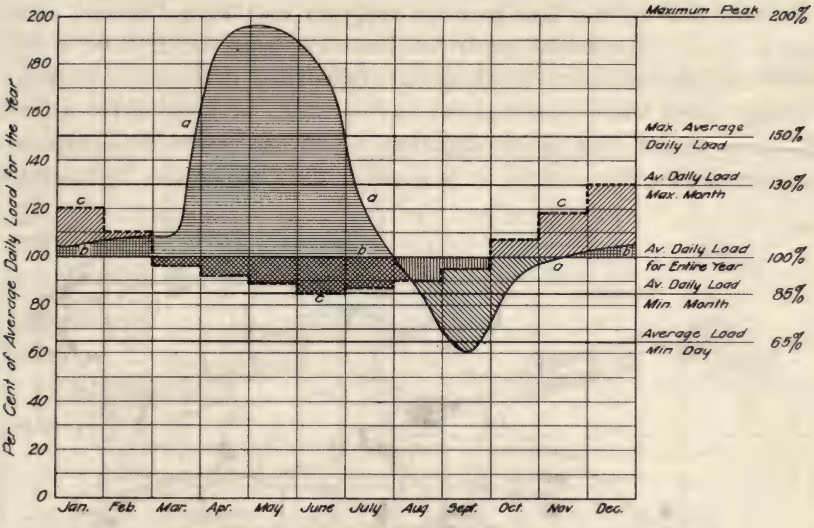
Fourth: The auxiliary power required to supply power during periods of more than average demand.

Fifth: The auxiliary power required to maintain the average load, but which will be saved on account of less than average demand for power.

Sixth: The average water not necessary to maintain average power but which is utilized on account of the average monthly load variation.

The effect of the variations of the daily load have been discussed in section 35, page 64, and illustrated by Figs. 37, 38 and 39, pages 65, 67 and 68.

From a study of this figure, it will be noted that there will be times when water is available, and is estimated in the table, when in fact it



Average water not used to maintain average power.

Water available to develop average power but which will probably be wasted (without storage) because less than average power will be used at that period.

Auxiliary power required to develop the average power during periods of deficient water supply.

Auxiliary power required to supply power during periods of more than average demand.

Auxiliary power required to maintain average load but which will be saved on account of less than average demand for power.

Average water not necessary to maintain average power but which is utilized to maintain average monthly load variation.

Fig. 86.—Mean Annual Hydrograph of a River and Its Relation to Load Requirements.

cannot be used on account of the irregularity of the proposed load, and there are also times when fuel has been estimated to supply auxiliary power that will not be needed for the same reason.

On the whole, it seems probable that more than the estimated amount of auxiliary power will actually be needed to maintain the average load.

77. **The Study of Comparative Hydrographs.**—It must be clearly understood that comparative hydrographs are of value only as the conditions are essentially similar on the various drainage areas compared.

Stream flow at the best is very irregular and varies greatly from year to year. The actual departure from the truth can best be understood and appreciated from an actual comparison of flows on adjacent drainage areas where observations have actually been made for a term of years. From such an investigation, which can be made as extended as desirable, the true weight to be given to the comparative hydrograph

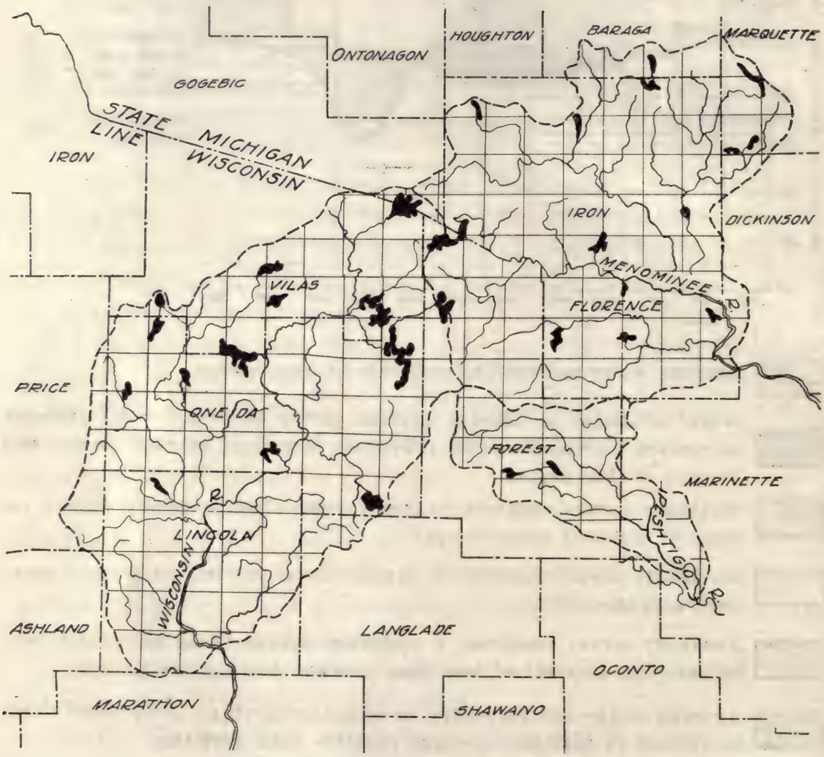


Fig. 87.—Drainage Areas of the Peshtigo, Upper Wisconsin and Menominee Rivers.

can best be judged. It is not believed that the actual variations from the truth, as shown by carefully selected comparative hydrographs, will be any greater than the flow variations which actually take place from a drainage area from year to year under the varying conditions of rainfall and climate. This method, therefore, is believed to be a scientific and systematic one for the consideration and discussion of probable variations in stream flow at any given point, if its limitations and the modifying influences known to exist on different drainage areas and

under different geographical, geological and meterological conditions are known and appreciated.

The estimates of possible power development on the Peshtigo River at High Falls, Wisconsin, were made when few gaugings were available

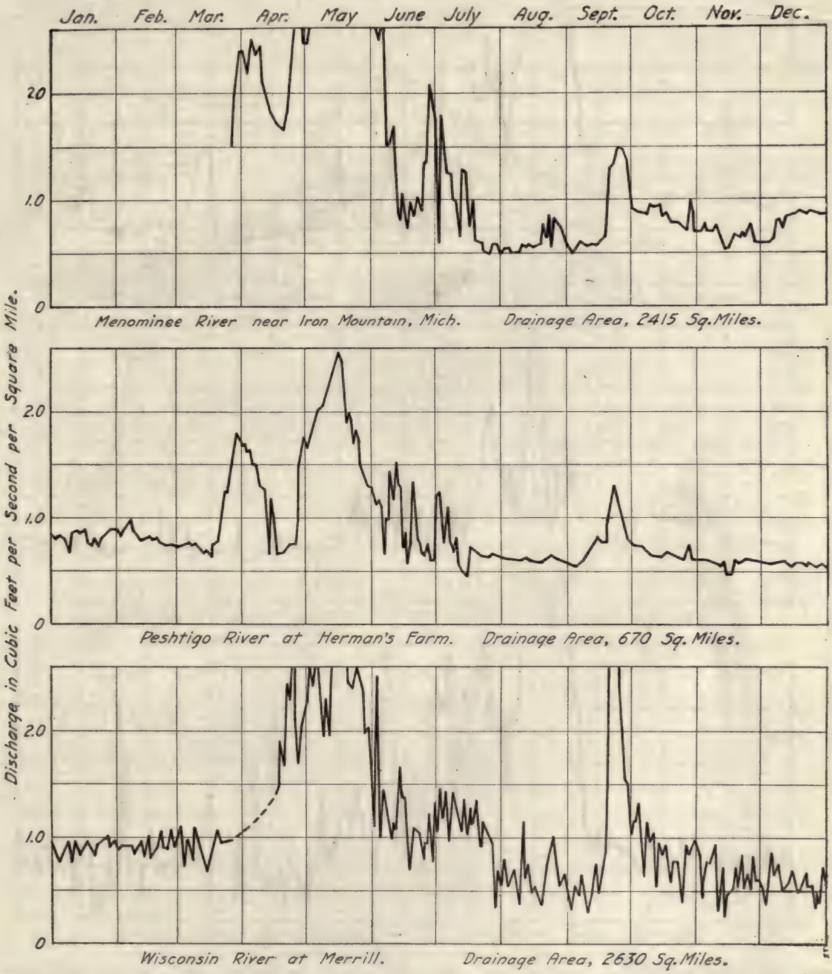


Fig. 88.—Comparative Discharge Hydrographs of Three Wisconsin Rivers for the Year 1907 (see page 162).

and were based upon the comparative hydrographs of the upper Menominee and upper Wisconsin rivers. The relative locations of the drainage areas considered are shown in Fig. 87, page 160. The observations made since the report was issued show that the flows of the

streams are fairly comparative as may be seen by reference to Fig. 88, page 161 and Fig. 89. That such comparisons may lead to erroneous conclusions, unless carefully chosen, may be seen by an ex-

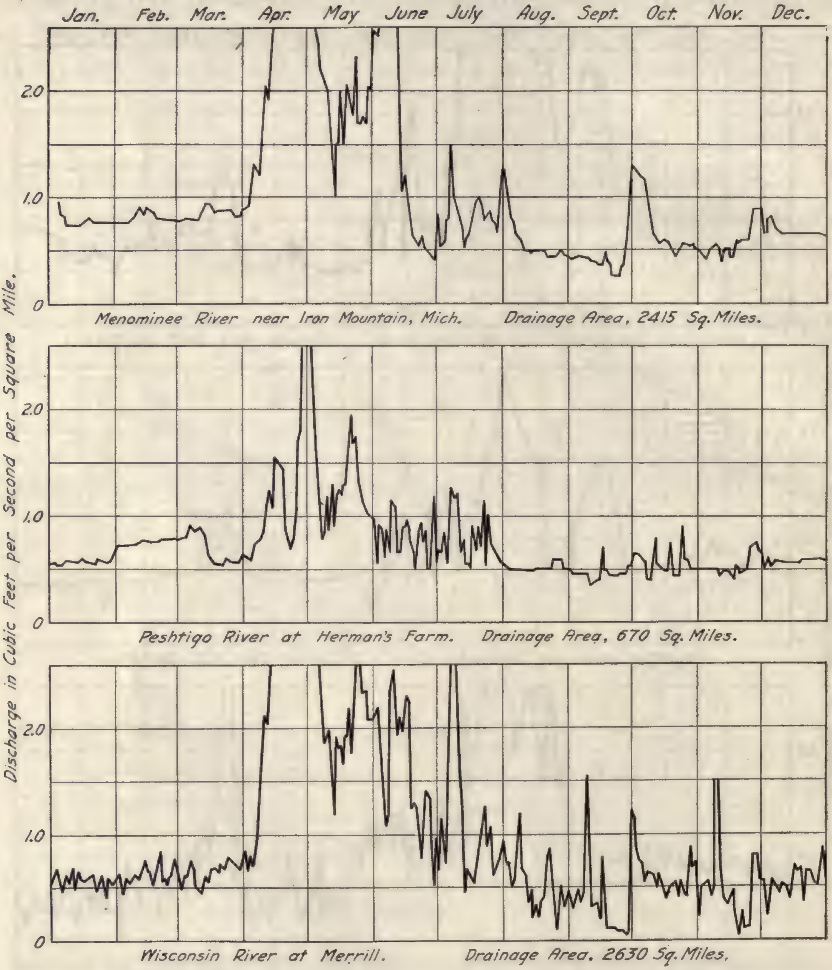


Fig. 89.—Comparative Discharge Hydrographs of Three Wisconsin Rivers.

amination of Fig. 91, page 164 and Fig. 92, page 165, which show the comparative chronological and duration hydrographs for various rivers of Michigan, the relative locations of which are shown in Fig. 90, page 163. From these hydrographs it may be seen that if the flow of the Thunder Bay River had been estimated on the basis of the actual

flow of the Au Sable River the estimate would have been twice too large. If the flow of the Au Sable River had been estimated on the basis of the actual flow of the Grand River at North Lansing, the estimated flow would have been only about one-third of the actual flow. Even the flows of the Grand River itself at North Lansing and Grand

Fig. 90.—Map Showing Location of Various Michigan Drainage Areas (see page 162).

Rapids differ considerably. *A study of comparative flow is always hazardous as a basis for any considerable investment and must be made with the greatest care.*

78. When no Hydrographs are Available.—In a new country where no observations are available either on the drainage area under consideration or on other areas adjacent thereto, the study of compara-

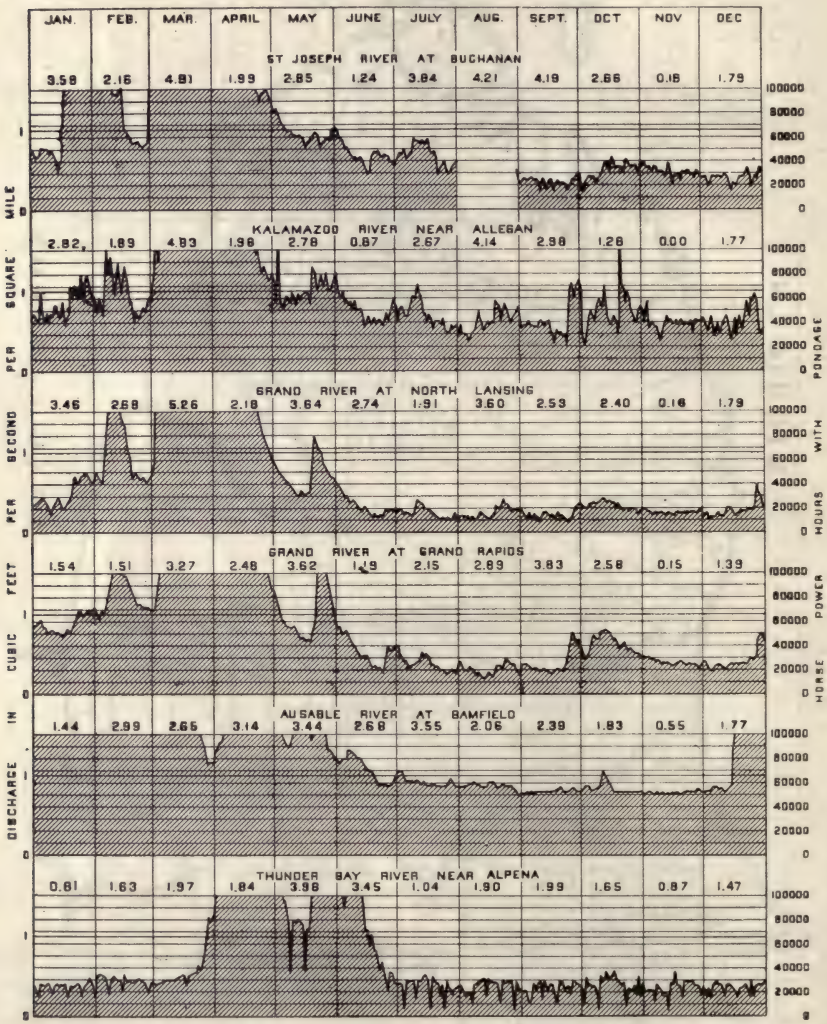


Fig. 91.—Comparative Hydrographs of Various Michigan Rivers for the Year 1904 (see page 162).

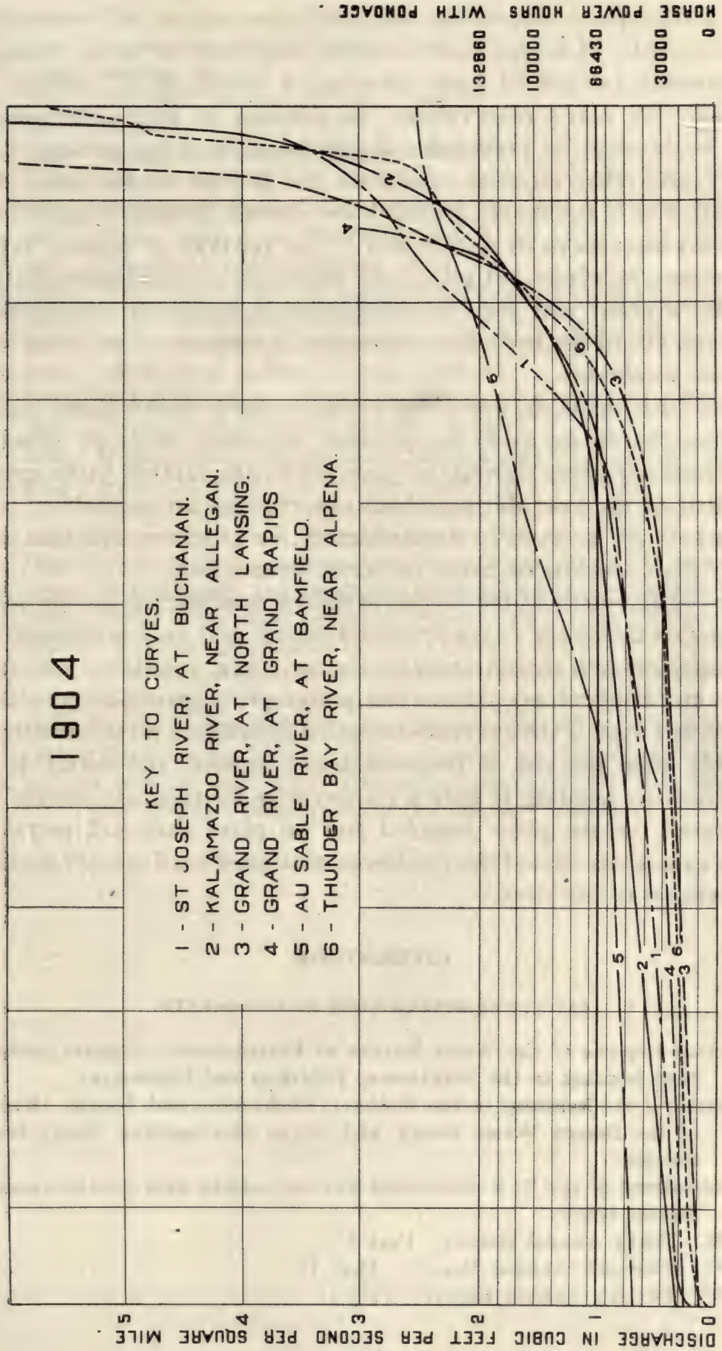


Fig. 92.—Duration Curves of Various Michigan Rivers (see page 162).

tive hydrographs is impossible and a different method of consideration must be used. If no data are available, time must be taken to acquire a reasonable amount of local information which should include not less than one year's observation. In addition to such observation a study as thorough as practicable should be made of the geology, topography, and other physical conditions that prevail on the water shed. Rainfall data is commonly available for a much greater range of time than the observations of stream flow. The relations of rainfall to run-off are very indefinite but give some indication as to whether the flow data for a given year may be regarded as a maximum, minimum or mean and afford the basis for a conservative estimate of probable average flow conditions.

From such relations, and from a single year's observations, conclusions may be drawn as to the probable variations from the observed flow which will occur during the years where the rainfall varies greatly from that of the year during which observations are available. Such conclusions are necessarily unsatisfactory, or at least much less satisfactory than conclusions based on actual stream flow.

The consideration of the best information available on any project is the basis on which the engineer should always rest his conclusions, and all relations which will throw light on the actual conditions should be given careful attention. If a water power plant must be immediately constructed upon a stream concerning which little or no information is available, then the risk is proportionately greater, and safety is obtained only by building in such a conservative manner that success will be assured for the plant installed and on plans that will permit of future extensions should the conditions that afterward develop warrant an extension of the same.

LITERATURE

RESULTS OF STREAM FLOW MEASUREMENTS

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2. Monthly Data Relating to the Sudbury, Cochituate, and Mystic. Reports of the Boston Water Board, and of the Metropolitan Water Board, Boston.

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3. 1888. Tenth Annual Report. Part I.
4. 1889. Eleventh Annual Report. Part II.
5. 1890. Twelfth Annual Report. Part II.

6. 1891. Thirteenth Annual Report. Part III.
7. 1892. Fourteenth Annual Report. Part II.
8. 1893. Bulletin No. 131.
9. 1894. Sixteenth Annual Report. Part II.
10. 1894. Bulletin No. 131.
11. 1895. Seventeenth Annual Report. Part II.
12. 1895. Bulletin No. 140.
13. 1896. Eighteenth Annual Report. Part IV.
14. 1896. Water Supply and Irrigation Paper, No. 11.
15. 1897. Nineteenth Annual Report. Part IV.
16. 1897. Water Supply and Irrigation Papers, Nos. 15 and 16.
17. 1898. Twentieth Annual Report. Part IV.
18. 1898. Water Supply and Irrigation Papers, Nos. 27 and 28.
19. 1899. Twenty-first Annual Report. Part IV.
20. 1899. Water Supply and Irrigation Papers, Nos. 35 to 39, inclusive.
21. 1900. Twenty-second Annual Report. Part IV.
22. 1900. Water Supply and Irrigation Papers, Nos. 47 to 52, inclusive.
23. 1901. Water Supply and Irrigation Papers, Nos. 65, 66 and 75.
24. 1902. Water Supply and Irrigation Papers, Nos. 81 to 85, inclusive.
25. 1903. Water Supply and Irrigation Papers, Nos. 97 to 100, inclusive.
26. 1904. Water Supply and Irrigation Papers, Nos. 124 to 135, inclusive.
27. 1905. Water Supply and Irrigation Papers, Nos. 165 to 178, inclusive.
28. 1906. Water Supply and Irrigation Papers, Nos. 201 to 214, inclusive.
29. 1907–8. Water Supply and Irrigation Papers, Nos. 241 to 252, inclusive.
30. 1909. Water Supply and Irrigation Papers, Nos. 261 to 272, inclusive.
31. 1910. Water Supply and Irrigation Papers, Nos. 281 to 292, inclusive.
32. 1911. Water Supply and Irrigation Papers, Nos. 301 to 312, inclusive.
33. 1912. Water Supply and Irrigation Papers, Nos. 321 to 332, inclusive.
34. Annual Reports and Water Resources Papers of the Dominion Water Power Branch, Department of the Interior, Canada.

CHAPTER VII

PONDAGE AND STORAGE

79. Effect of Pondage on Power.—The terms "Pondage" and "Storage" are quite similar in meaning, both having reference to the impounding of water for future use. The term pondage usually refers to the smaller ponds which permit of the impounding of the night flow for use during the working hours of day. Storage, on the other hand, is usually applied to the larger impounding reservoirs that enable a sufficient quantity of water to be stored to carry the plant, to some extent at least, through the dry season of the year. Between these limits every variation in capacity is of course possible.

In Chapter VI, Section 73, the effect of pondage on the power of a stream is briefly outlined and illustrated by hydrographs shown in Figs. 81, page 147 and 82, page 148. The pondage illustrated by these diagrams is sufficient to store the entire flow of the river during the parts of the day when the power is not in use and reserve it for those hours of the day when the power is needed. Such a condition can frequently be realized for the low flows during the dry seasons, but the capacity is seldom sufficient to store the larger flows, and if sufficient should be investigated in a different manner to be discussed later. These hydrographs should therefore be examined with these points in view.

In some water power installations practically no pondage is possible and the power of the stream must be utilized as it flows or otherwise it will be wasted. On continuous service, such as is sometimes required by cotton factories, paper mills, and electro-chemical works that run twenty-four hours per day, pondage is not so essential. With most power loads, such as are shown by the various load curves in Chapter III, the night load is small and the pondage of the night flow will frequently permit of more than doubling the power that can be otherwise utilized.

80. Effect of Limited Pondage on the Power Curve.—Frequently limited pondage only is possible and its influence on the possible power that can be generated must be carefully investigated. If power is to be used for a limited number of hours each day, the rate at which power can be used for this time without pondage will be the same as for the continuous power of the stream.

Such proportions of the otherwise unutilized flow of the stream as can be impounded during periods of light load can be added to the daily output. Thus, if power is used for twelve hours per day, and the night flow can be impounded and utilized during the day, the day power will be increased to double what it otherwise would be.

If power is used for only ten hours per day, with perfect pondage the day power will be increased to two and four-tenths of what it would otherwise be.

In twelve hours there are 43,200 seconds, and in each acre there are 43,560 square feet. It can therefore readily be remembered that for twelve hour pondage there must be practically as many acres one foot deep (or acre feet) in the pond as there are cubic feet per second to be impounded. For ten hour use and fourteen hour storage, the pond area must be increased by one-sixth above the capacity needed for twelve hour service. For example: In order to utilize the full flow of the Wisconsin River at Kilbourn in twelve hours (see Fig. 79, page 145), on the day of lowest flow (in August, 1904), a pondage of 3,000 acre feet would have been necessary, and, to utilize this full flow in ten working hours, would have required a pondage of about 3,500 acre feet.

Where the depth of pondage is considerable the effect of the variation in head on the power should receive careful consideration.

81. Power Hydrograph at Sterling, Illinois.—In 1903 an investigation was made of the probable effect, on the water power at Sterling, Illinois, of the proposed diversion of water for feeding the Illinois and Mississippi or "Hennepin" Canal.

The pondage formerly available at Sterling, by using eighteen inch flash boards on the dam, was about 42,000,000 cubic feet (almost 1,000 acre feet).

The diversion dam at Sterling has been constructed about one mile above the dam of the Sterling Hydraulic Company and has limited the available pondage to an area of about 5,000,000 square feet, and a pondage of about 7,000,000 cubic feet. This change has therefore caused a loss of pondage of about 35,000,000 cubic feet, which represents the night storage (i. e., the storage during the fourteen hours of night), of 700 cubic feet per second, which represents 980 hydraulic horse power for the ten hours of day. That is to say,—the loss of 35,000,000 cubic feet of storage capacity caused by the construction of the United States Government dam near the mouth of the Illinois and Mississippi Canal, has lost to the Sterling Hydraulic Company about 980 hydraulic horse power during such periods as the flow of the river is more than 840

cubic feet per second, and less than the capacity of the wheels installed (i. e., 4,450 cubic feet per second).

Figure 93, page 171, gives a graphical illustration of the effects of storage on the normal water power at Sterling and the loss resulting from the diminished storage. The lower flow line is the line of the normal hydraulic horse power of the Rock River for continuous (twenty-four hour) service. It also shows the total power available for ten-hour service without pondage. The flow line just above the line of normal power, and parallel thereto, shows the additional ten-hour power available from a pondage of 7,000,000 cubic feet. The upper flow line shows the ten-hour power made available by the storage of 42,000,000 cubic feet. The hatched area between lines two and three represents therefore the loss in ten-hour power which has been caused by the decrease in storage of 35,000,000 cubic feet.

From this diagram it will be noted that when the flow of the river is sufficient to supply the wheels, no loss would be occasioned by the decrease in pondage, and, as the flow approaches this point, the actual loss decreases. It should also be noted that when the flow of the river is less than 840 cubic feet per second (above the amount diverted by the canal) the total storage of 42,000,000 cubic feet is more than necessary to store the night flow, hence the loss at such times caused by the diminished pondage also decreases.

The approximate total loss of power for the year caused by the loss of 35,000,000 cubic feet of storage, as measured from this diagram, is 980 hydraulic horse power for, approximately, 250 ten-hour days.

82. Effect of Pondage on Other Power.—The pondage of water during the night naturally interferes with the normal flow of the stream and alters the regimen of the river at points below the point of pondage. The effect of such interference on other power, and the effect of other ponds on the plant contemplated, should be carefully considered.

The hydrographs of the Fox River (see Figs. 84 and 85, pages 152 and 153) are taken from observation by the government engineers at Rapide Croche, Wisconsin. Above this point are a number of water power dams. Many of the plants run twenty-four hours daily, but close down on Sundays. The effect of the Sunday shut-down on the stream flow is well shown in the hydrograph and is evident even during flood periods. It is to be noted that if a certain minimum power must be maintained at all times by a plant on the lower river, this interference in flow may create the necessity for an auxiliary plant, or the necessity of enlargement of the auxiliary plant otherwise required in order to maintain the required capacity at all times.

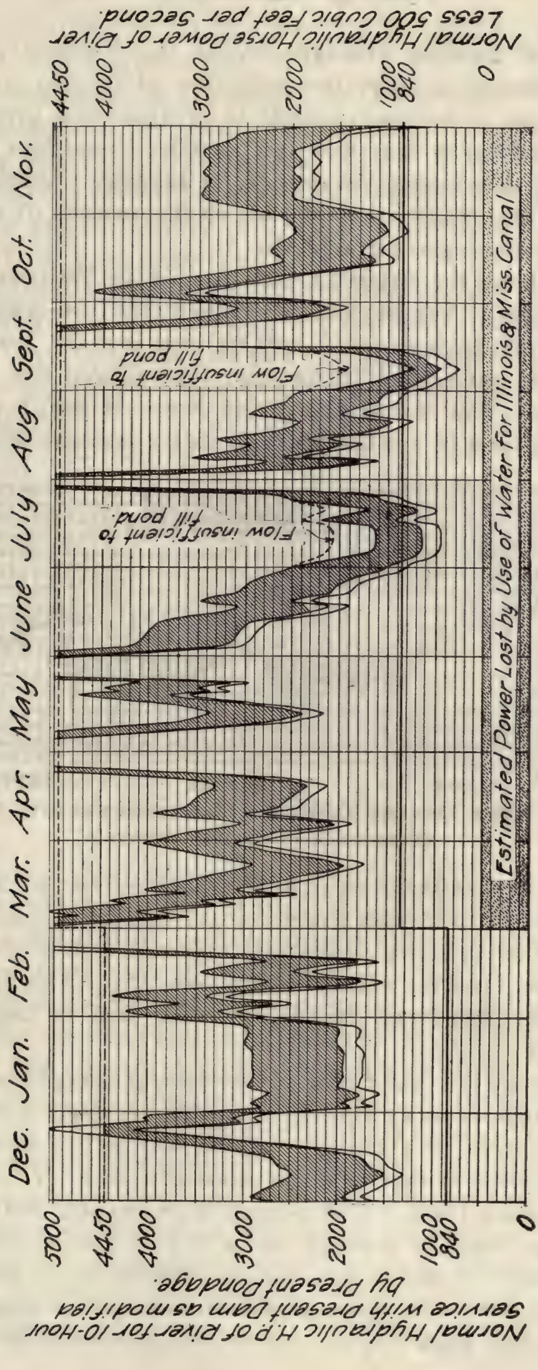

Fig. 93.—Power Hydrograph Showing Effect of Pondage on the Ten-hour Power of the Rock River at Sterling, Illinois (see page 170).

83. Effect of Limited Storage.—When the pondage available is more than sufficient to carry the night flow of the low water period over for day use, it becomes possible to equalize, to a greater or less extent, the variation in daily flow and to utilize excess flow to increase deficient flows, thus raising the quantity of available continuous power. The extent of this equalization depends on the quantity of storage and can readily be investigated graphically.

Figure 94 shows the estimated daily flow of the Wisconsin River at Kilbourn for July, August, and September (the low water period), 1904. From this hydrograph it will be seen that the lowest flow is 3,000 cubic feet per second. From Sec. 80 it is seen that in order to utilize the night flow during the twelve hours of day, a pondage of 3,000 acre feet must be available. With such a pondage the night flow

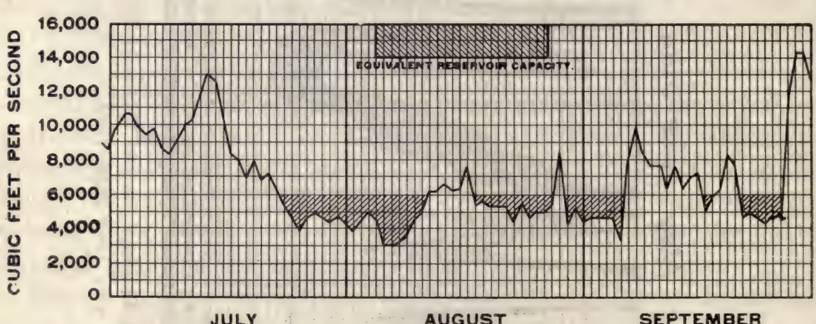

Fig. 94.—Low Water Flow at Kilbourn and Storage Capacity Necessary to Augment it to Various Amounts.

can ordinarly be distributed so as to be available either for twelve hour constant power or to furnish power for any equivalent load curve.

In Fig. 94 the horizontal spaces each represent a flow of 1,000 cubic feet per second, and the vertical spaces, one day. The area of each space therefore, represents 86,400,000 cubic feet, or approximately 2,000 acre-feet.

To increase the low water flow of the river to 4,000 second feet will require a storage capacity equivalent to that represented by approximately three spaces, or a storage of 6,000 acre feet in addition to the pondage, or a total storage of about 9,000 acre feet. To increase the flow to 5,000 second feet, a total storage of 28,000 acre feet in addition to the pondage would be required; and a flow of 6,000 second feet, will require a storage of 90,000 acre feet in addition to the pondage. In this latter case the conditions to September 6th must be considered, for the increased flow from August 12th to 17th is not sufficient to fill the

reservoir, although it will reduce the capacity required, as will also the increased flow of August 20th.

The reservoir capacity represented by 90,000 acre feet is shown on the diagram both by the curved hatched area above the flow-line and by the rectangular shaded area as well.

If the reservoir capacity is known, and its equivalent represented on the drawing, its effect on the hydrograph can readily be determined by trial (see also Fig. 97, page 176).

84. Effect of Large Storage.—When large storage is available, the daily flow of a stream can be equalized and its variations therefore becomes less important. In such cases the power of a plant depends on the average weekly or monthly flow of the stream and the possible storage capacity.

S. B. Hill, C. E., has suggested a method of discussing the effect of storage on the flow and power of a stream which is illustrated by Figs. 95 and 96. These hydrographs were prepared by the writer to illustrate a report on the probable power of a proposed hydraulic development in the south. Fig. 95 represents the mean monthly flow of the river in question for the years 1893 to 1906 inclusive. In this case the scale above the zero line shows both the mean monthly flow of the stream in cubic feet per second and the mean monthly power of the stream in horse power hours per day with the head available. The available storage is here 51,000 acre feet or 2,221,560,000 cubic feet. This storage is equivalent to a flow of 857 second feet for thirty days, or a storage of energy, with the available head, of about 5,000,000 H. P. hours.

The maximum daily continuous power (see *A-A*, Fig. 95) is determined by the effect of the driest year (viz. 1904) on the storage. The effect of the dry periods on the storage is shown by the incisions into the lower or storage line of the diagram. In the year 1904 the reservoir capacity would have been just exhausted in order to maintain the power during the low flows of September, October and November of that year. The amount of available continuous energy (i. e. the position of the line *A-A*) is determined by equalizing the deficiency in flow during the dry months with the total reservoir capacity.

It is important in the study of storage to see that in the intervening periods of excessive flow, such flows are sufficient to supply the deficiency occasioned by previous demands on the reservoir, otherwise the effect of one dry period must be considered in its relation to subsequent periods in determining the available continuous power (see Fig. 95, 1897 and 1898).

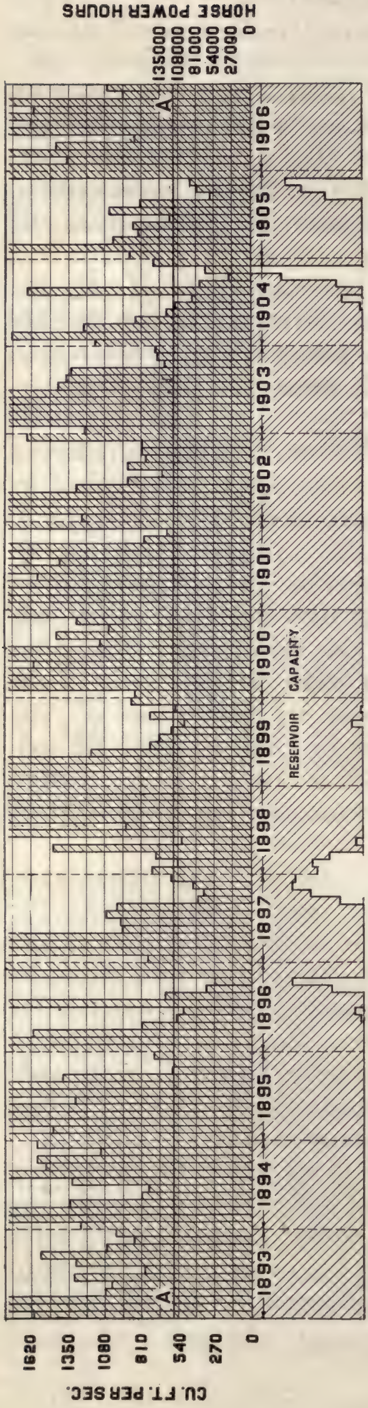

Fig. 95.—Mean Monthly Flow of a Southern River and the Effect of a Given Reservoir Capacity (see page 173).

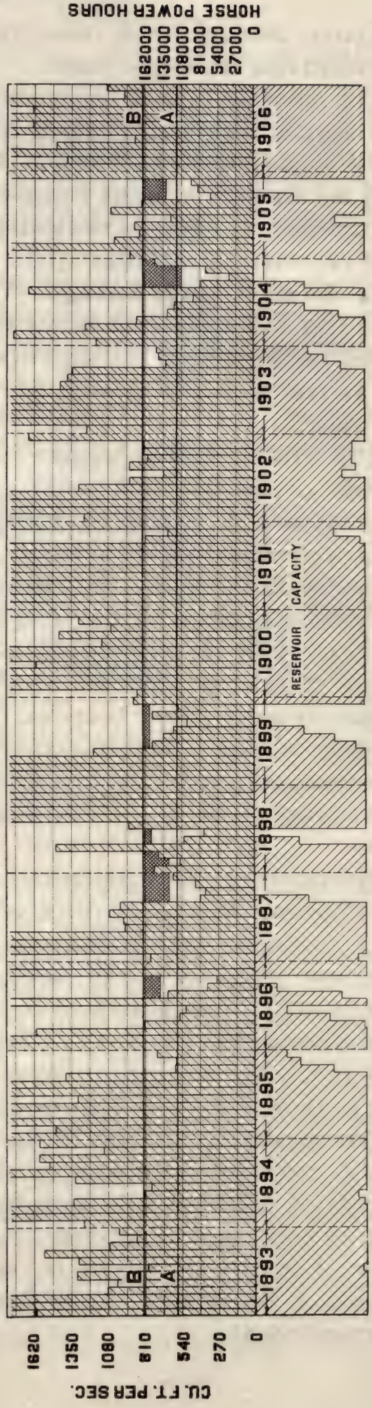

Fig. 96.—Amount of Auxiliary Power Necessary to Increase Output by 50,000 H. P. H. (see page 173).

The daily flow of this river for the year 1904 is shown by the hydrograph, Fig. 97, from which it will be seen that with pondage, but without storage, the available power of this stream would be limited to a minimum of 27,000 H. P. hours per day.

85. Effect of Auxiliary Power.—In order to maintain a continuous power greater than that due to the minimum flow of the stream plus the pondage, some source of auxiliary power must be available. If it is desired to increase the power of the stream represented in Fig. 95 by 50,000 H. P. hours per day, making the total horse power hours delivered 163,400 (represented by line *B-B,* Fig. 96), auxiliary power, as represented by the shaded areas on this diagram, would be needed. As at all other times water power would be available, the addition of steam auxiliary power would apparently be warranted. The size of the plant needed to furnish such excess power would depend on the method of power utilization. It is evident that during the dry periods in 1899, 1904 and 1905, if the water power was first used to its maximum, and the storage exhausted, an auxiliary plant would be needed of a capacity almost equal to the maximum demand on the plant, and that a plant of less capacity could be utilized satisfactorily only by operating it to a considerable capacity whenever a considerable draft began to be made on the storage. As the extent of the drought, or deficiency of water, could not be anticipated such a use of the auxiliary plant would require a greater expenditure of auxiliary horse power hours than is represented by the shaded areas in Fig. 96.

An investigation of the capacity and amount of auxiliary power needed, without pondage or storage, to maintain a given continuous power, can be readily made from the hydrograph of daily flow as shown by Figs. 98 and 99, page 176, which represent such a study of the Rock River at Sterling, Illinois, before the diversion of water for use in the Illinois and Mississippi canal, and the probable additional auxiliary power required to maintain the same power after such diversion.

86. Effect of Maximum Storage.—As the head increases the quantity of water needed to develop a given amount of power decreases, and storage becomes of much greater relative value. The storage of comparatively small quantities of water also becomes a more simple matter, but conditions which need little consideration with larger flows and lower heads, then become more important. In such cases, relatively large reservoir capacity sometimes becomes possible and only the questions of desirability and cost limit the extent to which such storage may be carried.

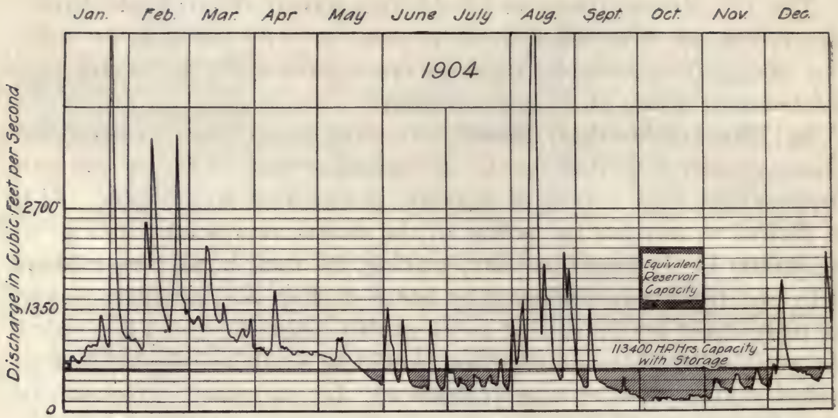

Fig. 97.—Discharge Hydrograph of Daily Flow of a Southern River (see page 175).

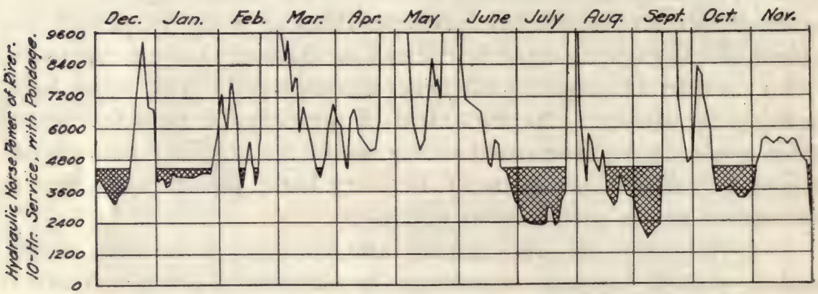

Fig. 98.—Power Hydrograph Showing Auxiliary Power Necessary to Maintain 4450 Ten-hour Horse Power at Sterling, Ill. (see page 176).

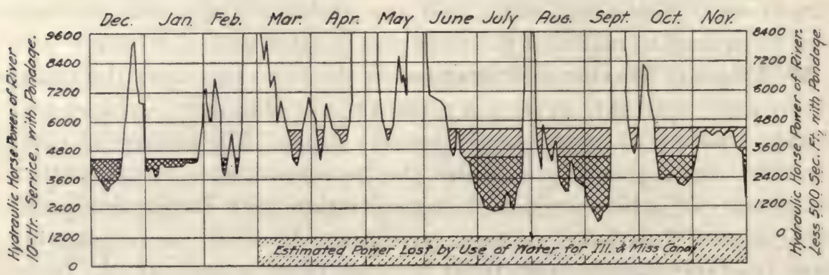

Fig. 99.—Power Hydrograph Showing Auxiliary Power Needed to Maintain Capacity of Wheels and Probable Increase Due to Diversion of Water for Illinois and Mississippi Canal (see page 176).

87. Calculations for Storage.—Rippl has outlined a method of computing storage which may occasionally be used to advantage under high head conditions, when it is desired to utilize the average flow of a series of dry months or years by extensive storage. This method consists in graphically representing the net yield of the stream during the period of low flow and from the curve of the net flow estimating the quantity of storage necessary for its full utilization.

The method suggested may be illustrated as follows:

From a study of the hydrographic conditions on the water shed for a considerable term of years, the period of extreme low flow is selected. For this period the observed or estimated flow of the stream for each month is reduced by the loss due to evaporation, seepage, etc. The remainder represents the net quantity of water available for power purposes. The summation of these monthly balances, added one to the other consecutively can be platted in a curve in which the abscissa of each point represents the total time from the beginning of the period; and the ordinate, the total quantity of water available during the same interval. The scale may represent inches on the drainage area, cubic feet, acre feet, or such other unit as may be desired. Such a curve is represented in Fig. 100 by the irregular curve *A-B-C-D-E-F*. The inclination of the curve at any point indicates the rate of the net flow at that particular time. When the curve is parallel to the horizontal axis, the flow at that time will just balance the losses caused by evaporation, seepage, etc. A negative inclination of the supply line shows that a loss from the reservoir is taking place.

In a similar manner the curve of consumption can be platted. For most purposes this can be considered a straight line as the variation in the use of power from season to season is a refinement not usually warranted, unless the uses to which the power is to be put at various times of the year are well established. In Fig. 100, page 178, a series of straight lines of consumption are drawn, representing the use of water at rates of 100 to 600 acre feet per day. These rates correspond essentially to rates of from 50 to 300 cubic feet per second.

The ordinate between the supply and any demand line represents the total surplus from the beginning of the period considered, and when inclination of the supply line is less than that of the demand line, the yield of the drainage area is less than the demand and a reservoir is necessary.

The deficiency occurring during dry periods is found by drawing lines parallel to the demand line, or lines, and tangent to the curve at the various summits of the supply curve, as at *B*.

The maximum deficiency in the supply, and the necessary capacity of the reservoir to maintain the demand during the period, is shown by the maximum ordinate drawn from the tangent to the curve itself. The period during which the reservoir would be drawn below the high water line is represented by the horizontal distance between the tangent point and the first point of intersection of the curve. If the tangent from any summit parallel to any demand line fails to intersect the

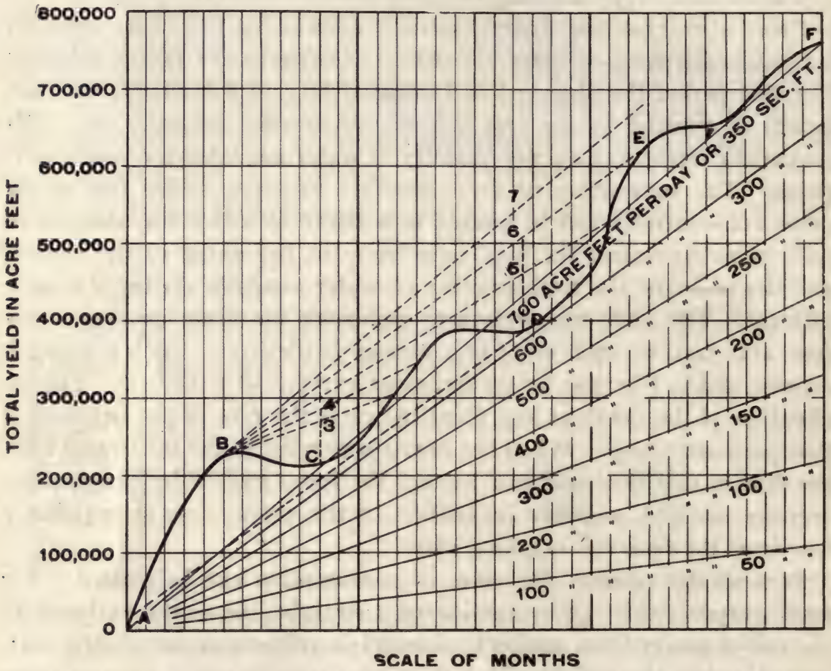

Fig. 100.—Diagram Illustrating Rippl Method of Calculating Storage.

curve, it indicates that, during that period, the supply is inadequate for the demand. To insure a full reservoir it is necessary that a parallel tangent drawn backward from the low points on the supply curve shall intersect the curve at some point below. For example: The line *B*-7, representing a daily consumption of 700 acre feet, does not again intersect the curve and is therefore (within the period represented by the diagram) beyond the capacity of the stream. The line *B*-6 intersects the curve at *E* and is the limit of the stream capacity. Such a consumption will be provided by a storage of about 150,000 acre feet as represented by the length of the line 6-*D*, and such a reservoir will be below the flow line for about twenty-two months during the dry period

illustrated in this diagram. That this reservoir will fill is shown by the intersection of the lower tangent *D-A* with the curve near *A*. The conditions necessary to maintain rates of 500, 400 and 300 second feet are shown respectively by the tangents *B*-5, *B*-4 and *B*-3, and the verticals 5-*D*, 4-*C* and 3-*C*.

If the amount of storage is known, and it is desired to ascertain the maximum demand that can be satisfied by such fixed capacity, the rate

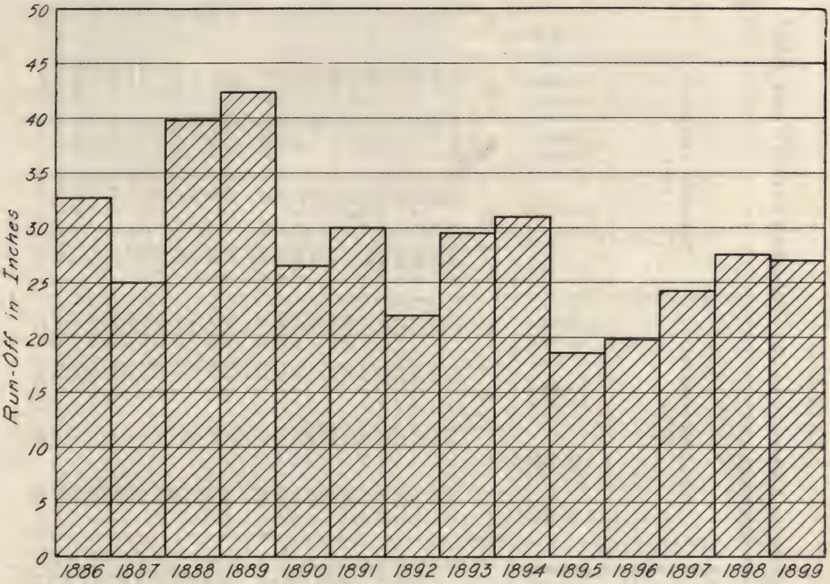

Fig. 101.—Diagram Showing Annual Run-off from Tohickon Creek.

is determined by drawing various tangent lines from the summits, having the maximum ordinates equal to the fixed storage.

88. Method of Storage Calculations.—The results of calculations, as outlined in Sec. 87 for various conditions of storage on Tohickon Creek, are shown in Table 17, page 180 and Fig. 102, page 182. Tohickon Creek is one of the possible sources of water supply which has been investigated by the city of Philadelphia for a considerable period. The observed monthly rainfalls and stream flows from the drainage area of this stream (in inches on the drainage area) are given in Tables 18 and 19. A five year period of low flow is found by inspection to run from December, 1893, to November, 1898, as shown by Fig. 101.

TABLE 17.

Calculation of Hydraulic Elements on the Tohickon Creek Water Shed.

Date.	Run-off in inches.	Rainfall in inches.	Evaporation and other losses.	3 — 4	With no reservoir area.	With 25% reservoir area.				With 50% reservoir area.				With 75% reservoir area.				With 100% reservoir area.
					Accum. sums of 2	75% of 2.	25% of 5.	Sum of 7 and 8.	Accum. sum of 9.	50% of 2.	50% of 5.	Sum of 11 and 12.	Accum. sum of 13	25% of 2.	75% of 5.	Sum of 15 and 16.	Accum. sum of 17	Accum. sum of 5.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
1894																		
December	3.10	3.10	—1.50	+1.60	3.10	2.33	+ .40	+2.73	2.73	1.55	+ .80	+2.35	2.35	.78	+1.20	1.98	1.98	1.60
January	.80	1.82	—1.00	+ .82	3.90	.60	+ .20	+ .80	3.53	.40	+ .40	+ .81	3.16	.20	+ .65	.85	2.83	2.42
February	3.79	3.96	—1.00	+2.96	7.69	2.84	+ .73	+3.58	7.11	1.89	+1.48	+3.37	6.53	.95	+2.22	3.17	6.00	5.38
March	3.08	1.65	—1.70	— .05	10.77	2.31	— .01	+2.30	9.41	1.54	— .02	+1.52	8.05	.77	— .04	.73	6.73	5.33
April	2.27	2.91	—3.00	— .09	13.04	1.70	— .02	+1.68	11.09	1.13	— .13	+1.08	9.13	.57	— .06	0.51	7.24	5.24
May	8.57	13.53	—4.50	+9.03	21.61	6.42	+2.26	+8.68	19.77	4.28	+4.51	+8.79	17.92	2.14	+6.77	8.91	16.15	14.27
June	.53	2.63	—5.50	—2.87	22.14	.39	— .72	— .33	19.44	.27	—1.44	—1.17	16.75	.13	—2.16	—2.03	14.12	11.40
July	.18	2.28	—6.00	—3.72	22.32	.13	— .93	— .80	18.64	.09	—1.86	—1.77	14.98	.04	—2.79	—2.75	11.37	7.68
August	.12	2.04	—5.50	—3.46	23.44	.09	— .86	— .77	17.87	.06	—1.73	—1.67	13.31	.03	—2.60	—2.57	8.80	4.22
September	3.33	9.44	—4.00	+5.44	25.77	2.50	+1.36	+3.86	21.73	1.66	+2.72	+4.38	17.69	.83	+4.08	4.91	13.71	9.66
October	2.10	5.18	—3.20	+1.98	27.87	1.58	+ .49	+2.07	23.80	1.05	+ .99	+2.04	19.73	.52	+1.47	1.99	15.70	11.64
November	2.67	3.00	—2.30	+ .70	30.54	2.00	+ .17	+2.17	25.97	1.34	+ .35	+1.69	21.42	.67	+ .52	1.19	16.89	12.34
1895																		
December	3.57	4.60	—1.50	+3.10	34.11	2.68	+ .77	+3.45	29.42	1.78	+1.55	+3.33	24.75	.89	+2.33	3.22	20.11	15.44
January	3.96	4.19	—1.00	+3.19	38.07	2.96	+ .79	+3.75	33.17	1.98	+1.59	+3.57	28.32	.99	+2.38	3.37	23.48	18.63
February	1.70	.96	—1.00	— .04	39.77	1.27	— .01	+1.26	34.43	.85	— .02	+ .83	29.15	.42	— .03	.38	23.86	18.59
March	5.37	3.11	—1.70	+1.41	45.14	4.03	+ .35	+4.38	38.81	2.68	+ .71	+3.39	32.54	1.34	+1.06	2.40	26.26	20.00
April	4.65	5.50	—3.00	+2.50	44.79	3.48	+ .62	+4.10	42.91	2.32	+1.25	+3.57	36.11	1.16	+1.87	3.03	29.29	22.50
May	.66	2.99	—4.50	—1.51	50.45	.49	+ .38	+ .87	43.78	.33	— .76	— .43	35.68	.16	—1.13	— .97	28.32	20.99
June	.27	4.49	—5.50	—1.01	50.72	.20	— .25	— .05	43.73	.14	— .50	— .36	35.32	.07	— .76	— .69	27.63	19.98
July	.81	3.53	—6.00	—2.47	51.53	.61	— .62	— .01	43.72	.41	—1.24	— .83	34.49	.20	—1.85	—1.65	25.98	17.51
August	.36	4.43	—5.50	—1.07	51.89	.27	— .27	.00	43.72	.18	— .53	— .35	34.14	.09	— .80	— .71	25.27	16.44
September	.04	.67	—4.00	—3.33	51.93	.03	— .84	— .81	42.91	.02	—1.67	—1.65	32.49	.01	—2.49	—2.48	22.79	13.11
October	.09	3.86	—3.20	+ .66	52.02	.07	+ .16	+ .23	43.14	.05	+ .33	+ .38	32.87	.02	+ .49	.51	23.30	13.77
November	.14	2.11	—2.30	— .19	52.16	.10	— .05	+ .05	43.19	.07	— .09	— .02	32.85	.04	— .16	— .12	23.18	13.58

1896																		
December	.66	2.51	—1.50	+1.01	52.82	.50	+ .25	+ .75	43.94	.33	+ .51	+ .84	33.69	.16	+ .76	+ .92	24.10	14.59
January	.54	1.18	—1.00	+ .18	53.36	.41	+ .04	+ .45	44.39	.27	+ .09	+ .36	34.65	.13	+ .13	+ .26	24.36	14.77
February	4.59	7.90	—1.00	+6.90	57.95	3.46	+1.72	+5.18	49.57	2.30	+3.45	+5.75	39.80	1.15	+5.18	+6.33	30.69	21.67
March	5.40	5.44	—1.70	+3.74	63.35	4.05	+ .94	+4.99	54.56	2.70	+1.87	+4.57	44.37	1.35	+2.80	+4.15	34.84	25.41
April	.73	1.48	—3.00	—1.52	64.08	.55	— .38	+ .17	54.73	.37	— .76	— .39	43.98	.18	—1.14	— .96	33.88	23.89
May	.30	3.18	—4.50	—1.32	64.38	.22	— .33	— .11	54.62	.15	— .66	— .51	43.47	.07	— .99	— .92	32.96	22.57
June	.18	4.07	—5.50	—1.43	64.56	.13	— .36	— .23	54.39	.09	— .72	— .63	42.84	.05	—1.07	—1.02	31.94	21.14
July	2.54	8.06	—6.00	+2.06	67.10	1.91	+ .52	+2.43	56.82	1.27	+1.03	+2.30	45.14	.63	+1.57	+2.20	34.14	23.60
August	.18	1.63	—5.50	—3.87	67.28	.13	— .97	— .84	55.98	.09	—1.93	—1.84	43.30	.04	—2.90	—2.86	31.28	19.33
September	1.12	5.83	—4.00	+1.83	68.40	.84	+ .46	+1.30	57.28	.56	+ .92	+1.48	44.78	.28	+1.38	+1.66	32.94	21.16
October	1.06	2.67	—3.20	— .53	69.46	.79	— .13	+ .66	57.94	.53	— .27	.26	45.04	.27	— .40	— .13	32.81	20.63
November	2.34	4.08	—2.30	+1.78	71.80	1.75	+ .44	+2.19	60.13	1.17	+ .89	+2.06	47.10	.58	+1.34	+1.92	34.73	22.41
1897																		
December	.81	.94	—1.50	— .56	72.61	.60	— .14	+ .46	60.59	.40	— .28	+ .12	47.22	1.00	— .42	+ .58	35.31	21.85
January	1.81	2.20	—1.00	+2.20	74.42	1.36	+ .55	+1.91	62.50	.90	+1.10	+2.00	49.22	.45	+1.65	+2.10	37.41	24.05
February	2.92	3.10	—1.00	+2.10	77.34	2.19	+ .52	+2.71	65.21	1.46	+1.05	+2.51	51.73	.73	+1.57	+2.30	39.71	26.15
March	2.19	2.46	—1.70	+ .76	79.53	1.64	+ .19	+1.83	65.14	1.09	+ .38	+1.47	53.20	.55	+ .57	+1.12	40.83	26.91
April	1.55	3.20	—3.00	+ .20	81.08	1.16	+ .05	+1.21	68.25	.78	+ .10	+ .88	54.08	.39	+ .15	+ .54	41.37	27.11
May	4.65	8.90	—4.50	+4.40	85.73	3.48	+1.10	+4.58	72.83	2.32	+2.20	+4.52	58.60	1.16	+3.30	+4.46	45.83	31.51
June	1.71	5.10	—5.50	— .40	87.44	1.28	— .10	+1.18	74.01	.86	— .20	+ .66	59.26	.43	— .30	+ .13	45.96	31.11
July	2.68	8.46	—6.00	+2.46	90.10	2.01	+ .62	+2.63	76.64	1.34	+1.23	+2.57	61.83	.67	+1.84	+2.51	48.47	33.57
August	.73	3.75	—5.50	—1.75	90.85	.55	— .44	+ .11	76.75	.36	— .87	— .51	61.32	.18	—1.31	—1.13	47.34	31.82
September	.12	1.92	—4.00	—2.08	90.97	.09	— .52	— .43	76.32	.06	—1.04	— .98	60.34	.03	—1.56	—1.53	45.81	29.74
October	.08	1.82	—3.20	—1.38	91.05	.06	— .35	— .29	76.03	.04	— .69	— .65	59.69	.02	—1.04	—1.02	44.79	28.36
November	1.78	5.03	—2.30	+2.73	92.83	1.34	+ .68	+2.02	78.05	.89	+1.37	+2.26	61.95	.44	+2.04	+2.48	47.27	31.99
1898																		
December	4.08	4.63	—1.50	+3.13	96.91	3.06	+ .78	+3.84	81.89	2.04	+1.57	+3.61	65.56	1.02	+2.34	+3.36	50.63	34.22
January	3.70	4.20	—1.00	+3.20	100.61	2.77	+ .80	+3.57	85.46	1.85	+1.60	+3.45	69.01	.92	+2.42	+3.32	53.95	37.42
February	4.05	3.38	—1.00	+2.38	104.66	3.03	+ .59	+3.62	89.08	2.03	+1.19	+3.22	72.23	1.01	+1.79	+2.80	56.75	39.80
March	1.83	2.84	—1.70	+1.14	106.49	1.37	+ .28	+1.65	90.73	.91	+ .57	+1.48	73.71	.46	+ .85	+1.31	58.06	40.94
April	2.50	3.73	—3.00	+ .73	108.99	1.87	+ .18	+2.05	92.78	1.25	+ .36	+1.61	75.32	.62	+ .55	+1.17	59.23	41.67
May	5.04	7.62	—4.50	+3.12	114.03	3.78	+ .78	+4.56	97.34	2.52	+1.56	+4.08	79.40	1.26	+2.34	+3.60	62.83	44.79
June	.19	.76	—5.50	—4.74	114.22	.14	—1.18	—1.04	96.30	.09	—2.37	—2.28	77.12	.05	—3.55	—3.50	59.33	40.05
July	.07	4.06	—6.00	—1.94	114.29	.05	— .49	— .44	95.86	.04	— .97	— .93	76.19	.02	—1.45	—1.43	57.90	38.11
August	.74	6.05	—5.50	—4.89	115.03	.55	—1.22	— .67	95.19	.37	—2.44	—2.07	74.12	.18	—3.65	—3.43	54.47	33.22
September	.08	2.03	—4.00	—1.97	115.11	.06	— .49	— .43	94.76	.04	— .99	— .95	73.17	.02	—1.48	—1.46	53.01	31.25
October	.61	5.21	—3.20	+2.01	115.72	.46	+ .50	+ .96	95.72	.30	+1.01	+1.31	74.48	.15	+1.50	+1.65	54.66	33.26
November	4.50	3.56	—2.30	+1.26	120.22	3.37	+ .31	+3.68	99.40	2.25	+ .63	+2.88	77.36	1.12	+ .94	+2.06	56.72	34.52

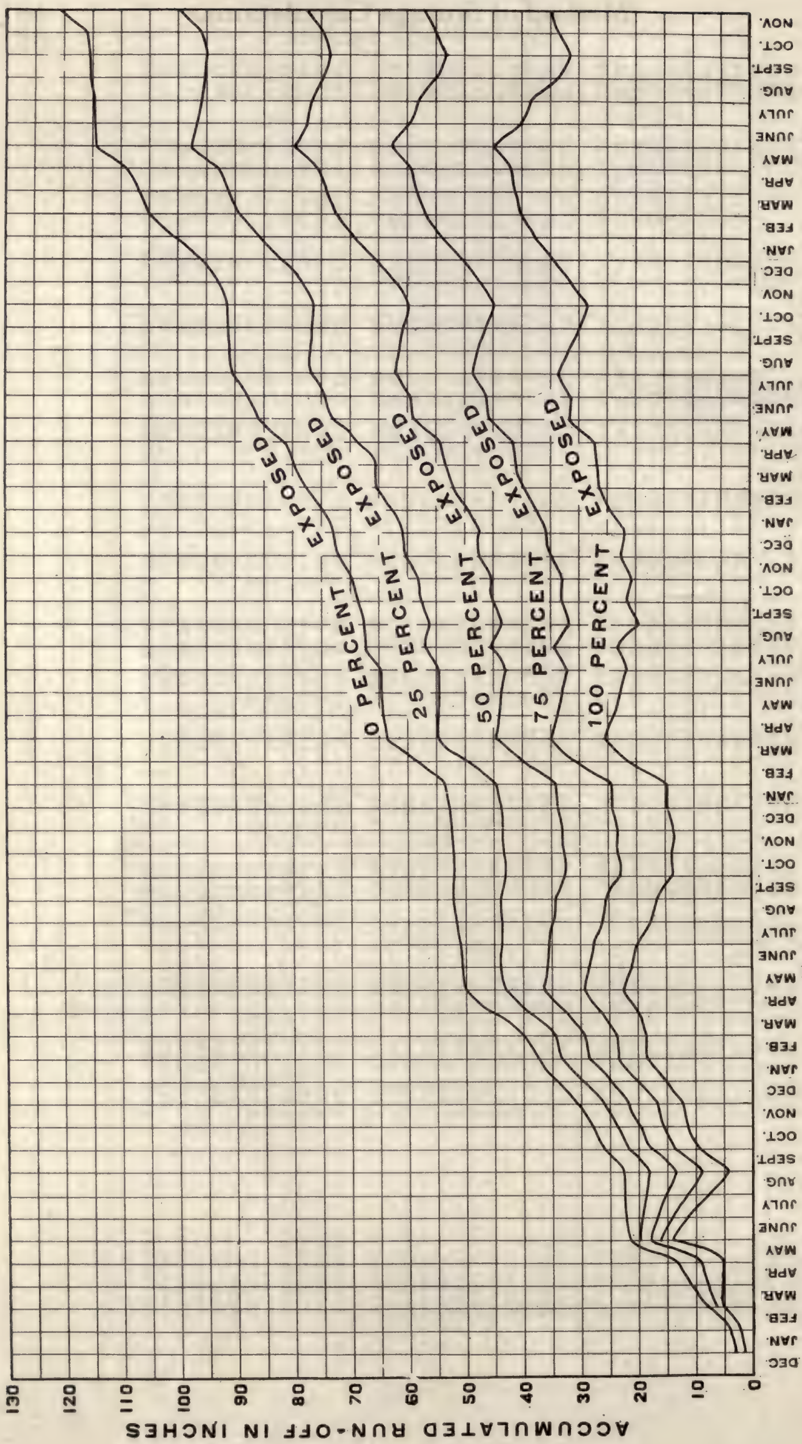

Fig. 102.—Mass Curve of Run-off of Tohickon Creek for Five Dry Years (see page —).

The calculations of the mass curves are based on the extreme variations in reservoir area of 0 to 100 per cent; that is, on the assumption that the reservoir may occupy from nothing to the entire drainage area.

TABLE 18.

Tohickon Creek—Monthly Rainfall in Inches.

Year.	Jan.	Feb.	Mar.	Apr.	May.	June.	July.	Aug.	Sept.	Oct.	Nov.	Dec.	Yearly Precipitation.
1886	4.15	6.01	4.76	3.42	7.14	4.53	5.47	1.08	1.30	2.59	5.16	3.83	49.45
1887	4.24	5.47	3.07	2.42	2.59	5.77	8.13	5.30	3.36	1.93	1.42	6.53	50.22
1888	5.31	4.34	5.23	4.08	3.03	1.69	3.20	8.07	8.32	4.06	3.66	4.35	55.34
1889	4.23	2.37	3.67	4.90	5.41	6.94	12.33	4.63	7.92	4.57	8.86	1.99	68.04
1890	2.82	4.73	6.77	2.48	6.30	3.93	5.81	5.75	2.98	6.31	1.07	2.75	51.60
1891	6.14	4.58	4.79	1.97	2.83	3.38	7.49	8.90	1.37	3.81	1.97	5.09	52.32
1892	5.49	1.23	4.13	1.95	5.55	3.20	4.27	3.76	2.91	.64	7.10	1.57	41.80
1893	2.96	5.88	2.46	4.96	4.98	4.05	2.10	8.67	3.20	3.72	4.37	3.17	50.52
1894	1.82	3.96	1.65	2.91	13.50	2.63	2.28	2.03	9.44	5.18	3.01	4.60	53.01
1895	4.19	.96	3.11	5.50	2.99	4.49	3.53	4.43	.68	3.86	2.11	2.57	38.24
1896	1.18	7.90	5.44	1.48	3.18	4.07	8.06	1.63	5.83	2.67	4.08	.94	46.46
1897	2.20	3.10	2.46	3.20	8.90	5.10	8.47	4.75	1.92	1.83	5.02	4.64	50.59
1898	4.19	3.38	2.84	3.73	7.62	.76	4.00	6.05	2.03	5.21	3.56	3.49	46.92
1899	3.68	4.75	6.60	2.19	2.23	2.74	3.29	5.05	6.70	1.39	2.55	2.34	43.51
Average	3.64	4.08	4.04	3.33	4.74	3.95	5.42	4.93	4.16	3.71	3.33	3.78	49.11

TABLE 19.

Tohickon Creek—Monthly Discharge in Inches on Drainage Area.

Year.	Jan.	Feb.	Mar.	Apr.	May.	June.	July.	Aug.	Sept.	Oct.	Nov.	Dec.	Yearly Discharge.
1886	4.36	9.19	4.28	4.75	3.43	1.41	.77	.09	.03	.05	1.91	2.38	32.65
1887	5.04	5.25	3.84	1.02	.93	1.21	1.63	1.96	.40	.25	.25	3.20	24.98
1888	6.38	6.72	6.27	4.28	.52	.15	.06	1.77	5.50	1.54	3.11	3.47	39.77
1889	4.38	1.51	3.86	2.88	1.70	2.29	6.41	3.75	3.40	2.33	7.97	1.92	42.40
1890	2.06	3.78	6.37	1.79	3.09	.75	.87	.92	1.22	3.54	.69	1.51	26.59
1891	6.15	5.68	5.03	1.58	.28	.17	.90	8.92	.94	.46	.63	4.27	30.01
1892	6.53	1.19	4.87	.84	2.05	.70	.51	.30	.19	.09	3.19	1.67	22.13
1893	2.22	6.64	4.54	3.22	3.79	.45	.10	1.56	.83	.60	2.62	3.10	29.67
1894	.80	3.80	3.09	2 28	8.58	.53	.19	.12	3.37	2.10	2.67	3.57	31.10
1895	3.95	1.70	5.37	4.65	.66	.27	.80	.37	.03	.09	.13	.67	18.69
1896	.54	4.59	5.48	.73	.30	.18	2.54	.19	1.12	1.06	2.34	.80	19.87
1897	1.81	2.92	2.19	1.55	4.63	1.71	2.68	.73	.12	.07	1.79	4.08	24.28
1898	3.70	4.05	1.83	2.50	5.04	.19	.07	.74	.08	.60	4.50	4.23	27.53
1899	4.72	5.56	8.99	1.57	.25	.07	.08	1.02	2.26	.19	1.02	1.28	27.01
Average	3.59	4.25	4.70	2.50	2.08	.76	1.15	1.19	1.36	1.20	1.89	2.89	27.58

The conditions on the reservoir area are those due to the equalization of the rainfall with the evaporation, seepage and other losses. The conditions on the balance of the water shed are given by the run-off and its summation.

Table 17 shows these calculations in detail and the mass curves drawn from columns 6, 10, 14, 18 and 19 are platted in Fig. 102. The maximum continuous power which could be maintained throughout

this period without storage is shown by the lowest slopes of the zero per cent. mass curve. The possible maximum development of the stream with various percentages of reservoir area can be determined by an analysis of the lower curves similar to that described in Sec. 87.

89. Analytical Methods.—Graphical methods of computation have been heretofore suggested as a means of investigating pondage and storage conditions. Such methods are believed to be advantageous in most cases on account of presenting visible evidence which can usually be more clearly understood than an abstract analysis.

Analytical methods for the consideration of these questions are usually obvious after the graphical methods discussed are understood, and such methods should usually be employed to check up the graphical deductions. Such methods may be illustrated by the following analysis of the effect of low water conditions on a proposed water power on a western river.

In this case daily gauge readings were available for about ten years, and the rainfall records were available for a considerably longer period.

From these records it appeared that the year 1905 was the driest year on record, and that the power available during the low water period of that year would have been equalled at least at all times during every year in the past twenty years, and with a probable like result in the future.

At the proposed plant each cubic foot per second, flowing during a day of twenty-four hours, would, at eighty per cent. efficiency, produce 3.63 continuous horse power. In order to develop 8,000 twenty-four hour horse power, it would be necessary, therefore, to have available a continuous flow of 2,200 second feet, while the minimum flow in 1905 was only 1,240 second feet. An examination of the gaugings shows that during the dry period of 1905 the water was deficient in quantity for sixty-eight days. The average flow for this period was 1,700 second feet, causing an average deficiency of 500 second feet. To impound sufficient water to maintain 2,200 second feet would require, therefore, a storage capacity of about 1,000 acre feet for each day of the dry period, or a total reservoir capacity of about 68,000 acre feet. Above the proposed dam site is a lake having an area of about sixty square miles or 38,400 acres. By raising the level of this lake two feet a storage of 76,800 acre feet would be attainable which, with careful manipulation would be sufficient to maintain the desired power.

If no storage were possible, and auxiliary power was to be established, the maximum capacity of the auxiliary plant would be determined by the day of lowest flow. During this day there was a de-

ficiency of 960 second feet, equivalent to about 3,500 H. P. The average deficiency for the period was 500 second feet, representing a necessary average of auxiliary power of 1,815 H. P., or 43,560 H. P. hours per day. The total auxiliary power for this period (sixty-eight days) would therefore be about 3,000,000 H. P. hours.

In the same manner the total amount of auxiliary power necessary during each year could be estimated and the interest and depreciation on the cost of the plant, plus the average annual operating expenses of the auxiliary plant, when considered in connection with similar elements of the water power installation, would furnish the basis for an estimate of the first cost and operating expenses of the combined plant to develop the required power.

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CHAPTER VIII

THE STUDY OF THE POWER OF A STREAM AS AFFECTED BY HEAD

90. Variations in Head.—In the previous chapter the graphical representation of stream flow has been considered. A method for the expression of the power resulting from the fluctuations of stream flow and under a constant head has also been shown. Experience shows, however, that such a condition seldom if ever occurs. In many cases where the available heads are considerable, the importance of the fluctuation in head is comparatively small, under which condition the diagrams already discussed are essentially correct and are satisfactory for the consideration of the varying power of the stream. In power developments under the low heads available in many rivers, the fluctuation in head has almost or quite as much influence on the continuous power that may be economically developed from a stream as the minimum flow of the stream itself.

The hydraulic gradient of a stream varies with the quantity of water flowing. At times of low water the fall available in almost every portion of its course is greater than is necessary to assure the flow between given points and frequent rapids result (see *R R*, Fig. 103) which are commonly the basis for water power developments. As the flow increases, however, a higher gradient and greater stream section is necessary in order to pass the greater quantity of water, and the rapids and small falls gradually become obscured (as shown by the medium water lines, Fig. 103 or disappear entirely under the larger flows (as shown by the higher water line, Fig. 103). Water power dams concentrate the fall of the river that is unnecessary to produce flow during conditions of low and moderate water (as shown in Fig. 104), and when the gradient of the water surface and the cross-section of the stream are increased to accommodate the larger flow, the fall at such dams is frequently greatly reduced (as shown by the medium water line in Fig. 104) or, during high water, the fall is largely or completely destroyed (as shown by the high water lines in the Figure), or at least is so reduced as to be almost or quite ineffective under practical water power conditions.

The cross-section of the river bed, its physical character and longitudinal slope, are the factors which determine the hydraulic gradient of

a stream under different flows. They are so variable in character and their detail condition is so difficult of determination that sufficient knowledge is seldom available, except possibly in the case of some artificial channels, to determine, with reasonable accuracy, the change of the surface gradient and cross-section of the water under various con-

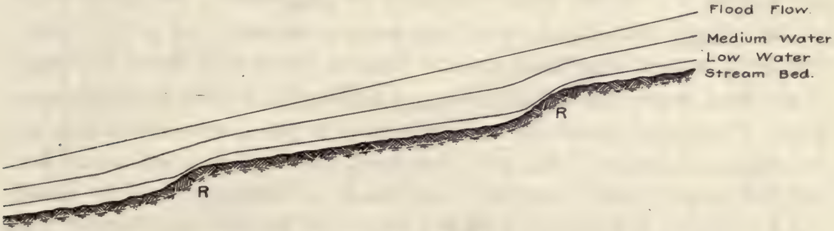


Fig. 103.—Hydraulic Gradients of a Stream Under Various Conditions of Flow.

ditions of flow. Where a power plant is to be installed, it is important to ascertain the relation of flow to head in order that the available power may be accurately determined. Where a river is in such condition as to make the determination of a discharge rating curve possible, either by direct river measurement at the point in question or by a comparison with the flow over weirs at some other point, such determina-

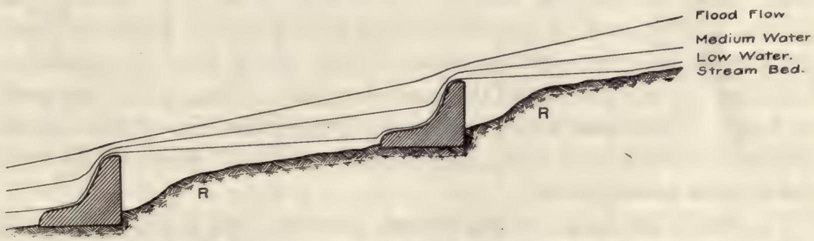


Fig. 104.—Hydraulic Gradients of the Same Stream After the Construction of Dam and Under Various Conditions of Flow.

tion should be carefully made, as such knowledge is of the utmost importance in considering the problem of continuous power.

91. The Tail Water Curve.—It will be readily seen that while the rating curve shows the relation between stream flow and river height prior to the construction of a dam, it will still represent the condition of flow below the dam after construction is completed. The water flowing over the dam will create a disturbed condition immediately below. If the velocity of the flow is partially checked or entirely destroyed, a heading-up of the water may result below the dam sufficient to give the velocity required to produce the flow in the river below, but

it will soon reach a normal condition similar to that which existed previous to the construction of the dam.

92. The Head Water Curve.—In Appendix C are shown (see Figure 436 and Figure 437) the discharge curves for weirs of various forms and the formulas representing them are also discussed in Chapter V. From these formulas or diagrams a discharge curve can be readily calculated, with reasonable exactness, for a dam with a certain form and length of crest. Such a curve will show the height of the head waters above the dam and under any assumed conditions of flow. From the rating curve of the river at the point considered, and the discharge curve of the weir proposed, the relative positions of head and tail waters under varying conditions of discharge can be readily and accurately determined, and if a weir is to be built to a certain fixed height, it will be seen that the head under any given conditions of flow may be thus determined.

93. Graphic Representation of Head.—Fig. 105 shows the rating curve of the Wisconsin River (see lower curve marked "Tail Water Curve") at Kilbourn. On this diagram has also been platted several discharge curves, two being for a weir of 300 feet in length and two for a weir of 350 feet in length. Both weir curves in the upper set are based on the assumption that the entire flow of water is passing over the weir. The crest of the dam is shown as raised to gauge nineteen, and the distance between the rating curve, which now represents the height of the tail water, and the weir discharge curves, which represent the height of the head water (with two different lengths of weir) under different conditions of flow, shows the heads that obtain at all times under these assumptions.

The entire discharge of the stream, however, will not pass over the dam except when the plant is entirely shut down, which will seldom be the case. The essential information which is desired therefore is the available head when the plant is in active operation. At the Kilbourn plant the discharge of the turbines installed under full head is 7,000 cubic feet per second; hence, with the plant in full operation, this quantity of water is passing through the wheels. Therefore in determining the relation between head water and tail water it must be considered that with a flow of 7,000 cubic feet per second, the water surface above the dam is at the elevation of its crest, no flow occurring over the spillway, and that only the flows greater than this amount pass over the dam. Another curve for each weir has therefore been added to the diagram in which the zero of the weir curves is platted from the point

where the line representing the height of the dam (elevation 19) intersects the line representing a discharge of 7,000 cubic feet per second. From this diagram (Fig. 105) it will be seen that other heads, shown in Table 20, will obtain under various conditions of flow.

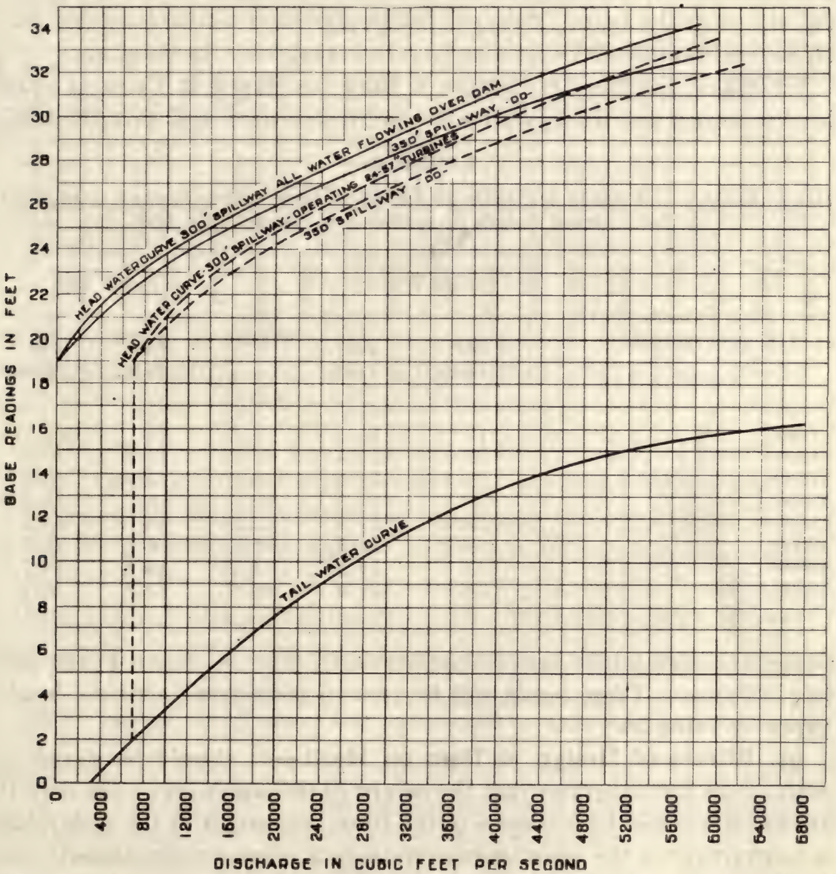


Fig. 105.—Showing Head at the Kilbourn Dam Under Various Conditions of Flow.

It will readily be seen that the line representing the height of the dam is not essential and that the curves may be platted relative to each other, leaving the height of the dam out of the question entirely and indeterminate. A curve constructed on this basis but otherwise drawn in the same manner as in Fig. 105, is shown in Fig. 106, page 191. In Fig. 106, wherever the weir or head water curves pass above the tail water curve,

it shows that an increase in the head will result under the corresponding condition of flow and wherever they pass below such curve, it shows that a decrease in the head will result under the corresponding condition of flow, the amount of which is clearly shown by the scale of the diagram. Consequently, having given the height of the dam above tail water at the point of no discharge, the head available under any other condition can be immediately determined from the diagram.

From this diagram the changes in head (as shown in Table 21) can be determined and these, with a seventeen foot dam, will give the total

TABLE 20.

Gauge heights and heads available at Kilbourn Dam under various conditions of flow, with a length of spillway of 300 and 350 feet.

Flow in cubic feet per second.	Head Water		Tail Water.	Head with	
	300 ft. dam.	350 ft. dam.		300 ft. dam.	350 ft. dam.
7000	19	19	2	17	17
14000	22.9	22.3	5.1	17.8	17.2
21000	25.2	24.6	8	17.2	16.6
28000	27	26.2	10.3	16.7	15.9
35000	28.5	27.7	12.2	16.5	15.5
42000	30.2	29.3	13.6	16.6	15.7
49000	31.5	30.4	14.7	16.8	15.7
56000	32.7	31.6	15.6	17.1	15.8

heads available under various conditions of flow as shown in the last two columns. These heads will be seen to correspond with the heads given in Table 20.

94. Effects of Design of Dam on Head.—It should be noted in both of the last diagrams that the height of the water above the dam is readily controlled by a change in the form and length of the weir; that a contraction in the weir length produces a corresponding rise in the head waters as the flow increases, while the lengthening of the weir will reduce the height of the head water under all conditions of flow. The physical conditions relative to overflow above the dam will control the point to which the head waters may be permitted to rise and will modify the length and the construction of the dam. Where the overflow must be limited, the waters, during flood times, must be controlled either by a sufficient length of spillway or by a temporary or permanent reduction in the height of the dam such as the removal of flash boards, the opening of gates, or by some form of movable dam.

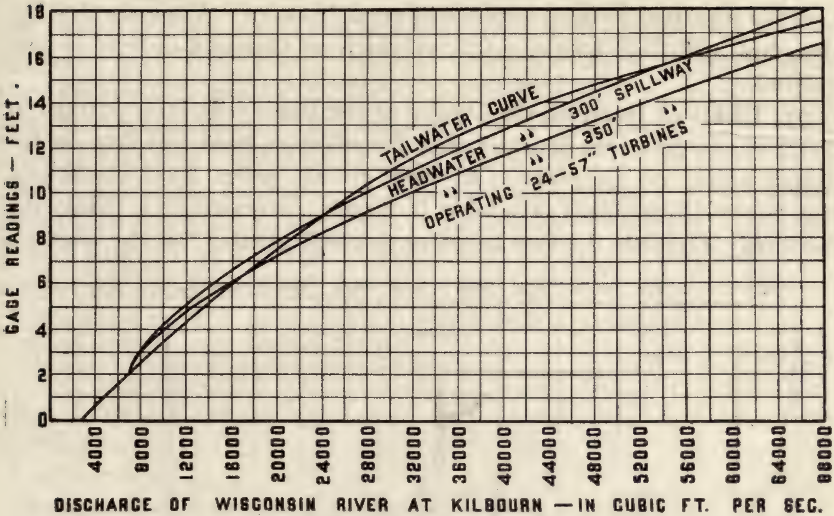


Fig. 106.—Showing Change in Head at Kilbourn Dam Under Various Condition of Flow (see page 189).

Having determined the head available at all conditions of river flow, the hydrograph, as previously shown, may be modified to show the actual power of the river under the varying conditions of flow. The vertical scale, in this case, instead of being uniform must be variable as the head varies. Fig. 107, page 192, shows graphically the variation in the continuous theoretical power of the river taking into considera-

TABLE 21.

Changes in head at Kilbourn Dam with lengths of crest 300 and 350 feet and under various conditions of flow with resulting total available head with 17 ft. dam.

Flow in cubic feet per second.	Changes in Head with		Total Head with	
	300 ft. dam.	350 ft. dam.	300 ft. dam.	350 ft. dam.
7000	0	0	17	17
14000	+ .8	+ .2	17.8	17.2
21000	+ .2	— .4	17.2	16.6
28000	— .3	—1.1	16.7	15.9
35000	— .5	—1.5	16.5	15.5
42000	— .4	—1.3	16.6	15.7
49000	— .2	—1.3	16.8	15.7
56000	+ .1	—1.2	17.1	15.8

tion the variation in head which will actually occur. Compare this hydrograph with Fig. 80, page 146, in which no variation in head is considered.

95. Effect of Head on the Power of the Plant.—It is important at this point to take into consideration the effect of head and flow on the actual power of the plant. In most rivers, under flood conditions, the power theoretically available is largely increased, for, while the head may diminish, the flow becomes so much greater that the effect of

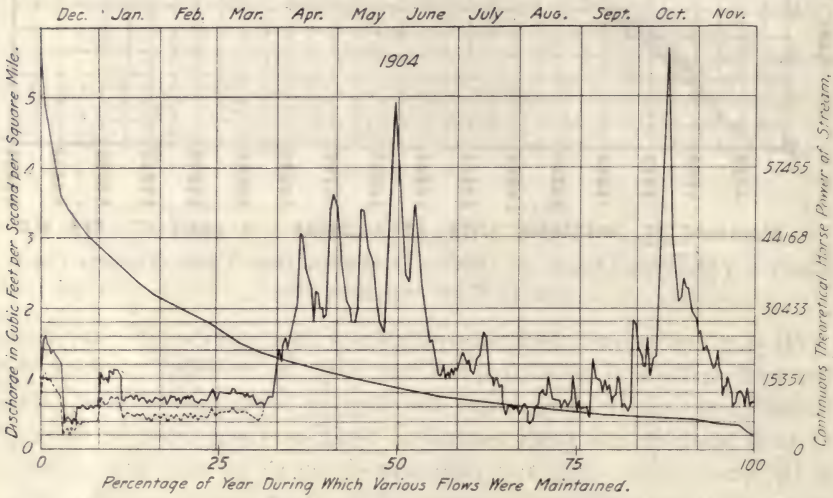


Fig. 107.—Power Hydrograph Showing Continuous (24 Hour) Theoretical Horse Power at Kilbourn, Wis., With Actual Head (see page 191).

head on the theoretical power is more than offset thereby. Practically, however, the conditions of head under which a given water wheel will operate satisfactorily (i. e. at a fixed speed) are limited, and, while the theoretical power of the river may radically increase, the power of the plant installed under such conditions will often seriously decrease, and under extreme conditions may cease entirely. The discharging capacity of any opening is directly proportional to the square root of the head, and the water wheel, or water wheels, simply offers a particular form of opening, or openings, and operates essentially under this general law. With a fixed efficiency, therefore, the power which may be developed by a water wheel is in direct proportion to its discharging capacity and to the available head. Hence, the power of the wheel decreases as the product of these two factors, and therefore the power available under conditions of high flow and small head are much

less than where the head is large and the total flow of the river is less. The only way, therefore, to take advantage of the large increase in theoretical power during the high water conditions is to install a surplus of machinery above that required for the condition of average water. This may sometimes be done to advantage, but its extent soon reaches a practical limitation on account of the expense. It often becomes desirable to take care of such extraordinary condition by the use of supplemental or auxiliary power. Such power can usually also be applied during conditions of low water flow when the power is limited by the other extreme of insufficient water under maximum head.

In considering the effect of head on the power of a plant, it is necessary to understand that water wheels are almost invariably selected to run at a certain definite speed for a given power plant and cannot be used satisfactorily unless this speed can be maintained. Also that any wheel will give its best efficiency at a fixed speed only under limited changes in head. If the head changes radically, the efficiency changes as well and this effect becomes more serious under a reduction in head. As the head is reduced, the discharging capacity of the wheel and its efficiency is also rapidly reduced so that the power of the wheel decreases more rapidly than the reduction in the discharging capacity would indicate. When the reduction of head reaches a certain point the wheel is able to simply maintain its speed without developing power, and when the head falls below that point, the speed can no longer be maintained. It is therefore plain that when the head of a stream varies greatly, it becomes an important and difficult matter to select wheels which will operate satisfactorily under such variations, and, when the variations become too great, it may be practically or financially impossible to do so. This subject is discussed at length in Chapters XI and XIII, but is called to the attention of the engineer as an important matter in connection with the study of head.

96. Graphical Investigation of the Relations of Power, Head and Flow.—The relation of head and flow to the horse power of any stream on which a dam has been constructed, may be graphically investigated and determined by a diagram similar to Fig. 108, page 194. On this diagram are platted hyperbolic lines marked "horse power curves" which show the relation of horse power to head and flow within the probable limits of the conditions at Three Rivers, Michigan. These lines are drawn to represent the actual horse power of a stream under limited variations in head and flow and on the basis of a plant efficiency of seventy-five per cent. These heads, which actually obtain at the

Three Rivers dam, were observed under three conditions of flow, and these observations were platted on the diagram at *e e e* and a curve was drawn through them. From the intersection of this curve with the

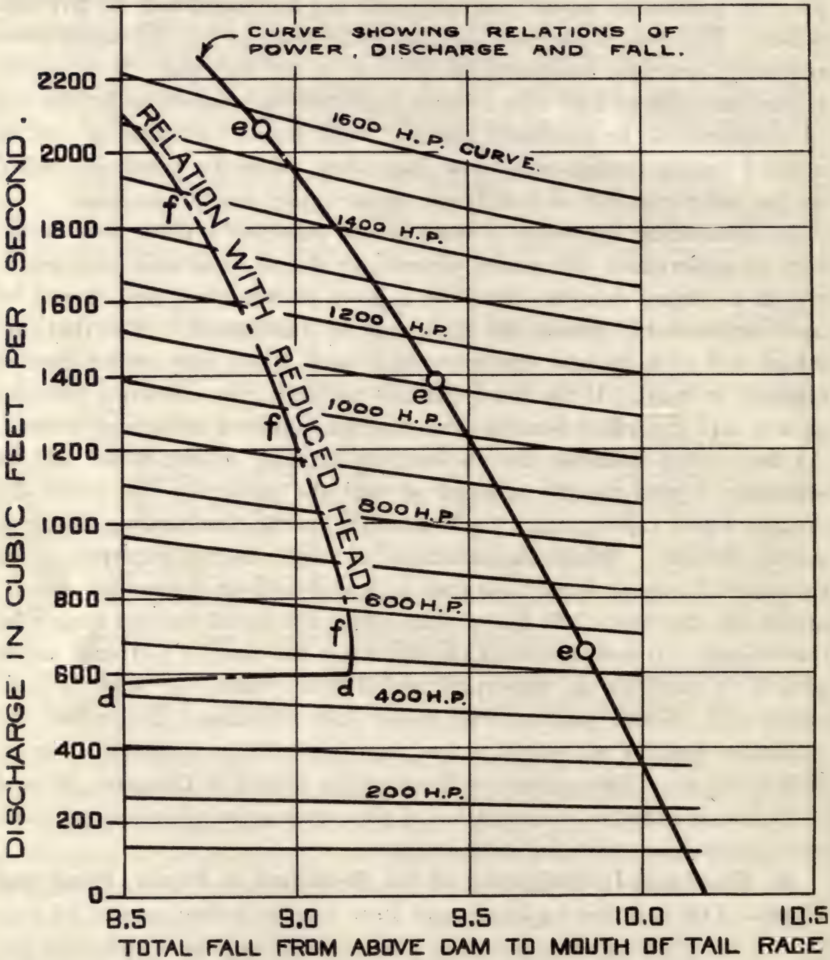


Fig. 108.—Graphical Study of Head (see page 193).

horse power curves, the actual power of the river available under the actual variations of head and flow, is determined. These measurements were taken with all of the water passing over the dam.

Let us assume that it is desired to investigate the effect of an installation of wheels, using 600 cubic feet per second, under a nine foot head. Under these conditions part of the water will pass through the turbines

instead of over the crest of the dam, the available head will therefore be somewhat reduced, and the power curve of the river, under these new conditions, is shown on the diagram by the curve *f f f*. This curve was platted from the curve *e e e* by computing the amount the head on the crest of the dam would be lowered at different stages of the river by diverting through the wheels the quantity of water which they will pass under the reduced head. The actual power of the river at different heads and under these conditions is shown by the intersection of the line *f f f* with the horse power curve, and the actual power of the proposed plant under various conditions of flow is obtained by projecting the point of intersection of the discharge line with the line *f f f* on the turbine discharge line *d d*.

Thus, with a flow of 600 cubic feet per second, the power of the plant would be about 470 H. P., while, with a flow of 2,100 feet per second, the power of the plant would decrease to about 420 H. P. At discharges below 600 cubic feet per second the head would drop rapidly unless a portion of the installation was shut down.

97. **Graphical Study of Power at Kilbourn.**—A more detailed study of head in connection with the conditions at Kilbourn, Wisconsin, is illustrated by Figs. 109, page 196 and 110, page 200. In Fig. 109, the theoretical horse power of any stream resulting from any variation between head and flow is shown by the hyperbolic curves, marked 10,000 H. P., etc. These curves are perfectly general and depend only upon the scales of river flow and "head" shown at the left and at the bottom of the diagram. These curves give the relation between head and river discharge to produce the theoretical horse power marked on each curve.

The curve marked "Discharge—24–57″ Turbines" is general and shows the total discharge of 24–57 inch turbines in cubic feet per second for the various values of head given on the scale at the bottom of the diagram. Fig. 105, page 189, previously considered, shows the relation between discharge of the river and height of head and tail water.

The curve marked "Height of Crest above Tail Water" was obtained by subtracting the height of tail water at various stages of the river as given by the rating curve of the river, from the height to which the spillway of the dam is to be constructed, and platting the values thus obtained in their correct position on the diagram. The dam here considered is seventeen feet in height above average water, and its crest corresponds to the elevation of nineteen feet on the gauge. Thus in determining this curve, the tail water gauge height (see tail water curve,

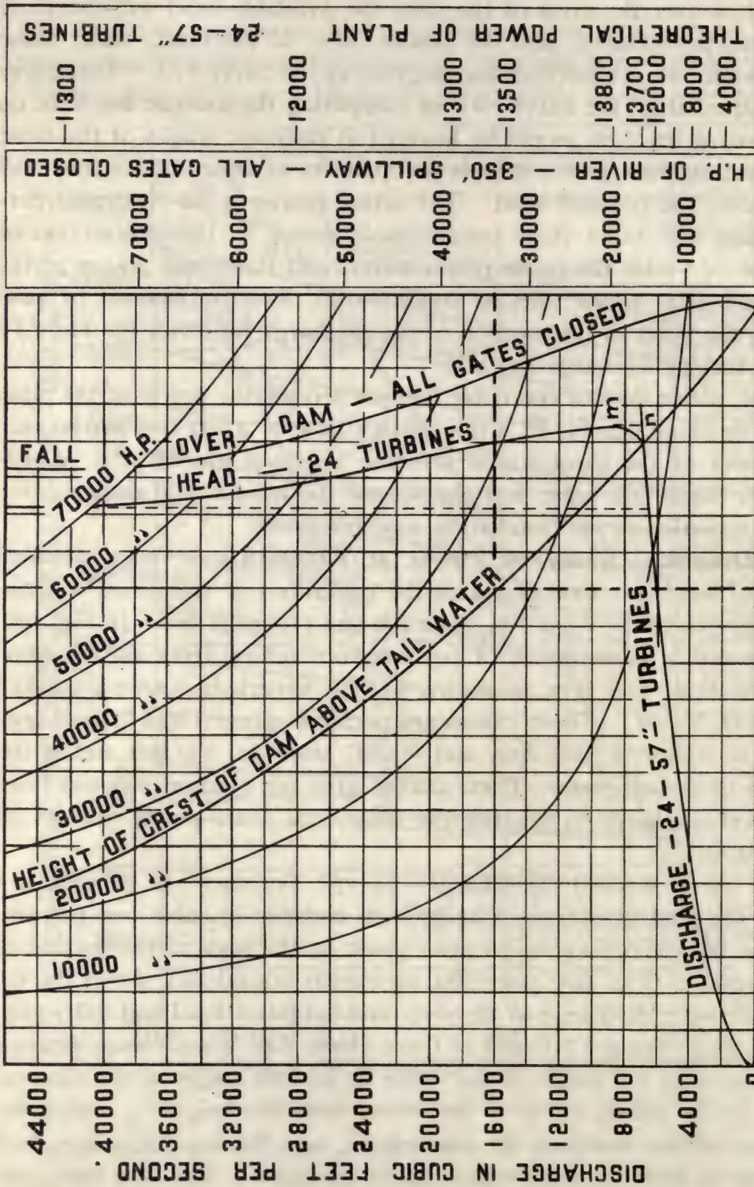


Fig. 109.—Graphical Analysis of Relation of Power, Head and Flow at Kilbourn Plant of Southern Wisconsin Power Co. (see page 195).

Fig. 105) was subtracted from nineteen, and this value platted on the scale of "head" at the bottom of the diagram at the given value of the discharge.

The curve marked "Fall over Dam, All Gates Closed" was constructed by laying off as abscissas on the scale of "head," the values of actual head as given by the vertical intercept between the tail water curve and head water curve, with all water flowing over the dam with 350 foot spillway (Fig. 105). This curve then gives the distance between head water and tail water for various quantities of river discharge. The horizontal intercept between this curve and the curve marked "Height of Crest above Tail Water," gives the depth on the crest of the dam with 350 foot spillway for various rates of river flow when the whole discharge is passing over the dam. Thus for any given river discharge, the total fall can be obtained by finding the intersection of the line representing the value of the discharge (for instance, 16,000 cubic feet per second), with the curve marked "Fall over Dam, All Gates Closed," giving in this case 18.8 feet. The theoretical horse power of the river under the same conditions is given by interpolating between the hyperbolic horse power curves (in the assumed case equals 34,000 horse power). This is also given by the first scale on the right hand side of the diagram.

The curve marked "Head—24 Turbines" shows the head acting on the turbines for various values of river discharge. The head acting on the turbines is less than the fall over the dam with all gates closed, due to the additional effect of the turbines themselves in drawing down the head water, while the tail water remains at its normal value, corresponding to the total flow of the river. Points on this curve are given by the values of the intercepts between the tail water curve and the head water curve, for the dam with 350 foot spillway, operating 24–57″ turbines. An examination of this curve in connection with the turbine discharge curve shows that the head on the turbines increases with the discharge of the river as the abscissas to a parabola with horizontal axis and origin at zero, until the height of crest curve is intersected, beyond which the head increases but slightly, and afterward reduces with further increase in river flow.

98. Power of the Kilbourn Wheels Under Variations in Flow.—When the gates to the turbines are open a less quantity of water will flow over the dam and the head on the crest will therefore be diminished. The amount of water which will pass through the proposed installation under various heads, is shown by the curve marked "Discharge 24–57″ turbines." The intersection of this curve, with the dis-

charge lines, at all points to the left of the curve marked "Height of Crest of Dam above Tail Water" indicates that such flows will pass through the wheels at the head indicated by the point of intersection. The practical limit of the turbine capacity is the discharge indicated by the point of intersection of the turbine discharge curve with the "Height of Crest of Dam above Tail Water." It will be noted that this intersection shows a maximum discharge of 7,000 cubic feet per second under a head of seventeen feet. A further increase in the discharge of the river up to 8,700 cubic feet per second, causes an increase in the head, which is found by following upward the curve marked "Head 24 turbines" to the point *m* where a maximum head is indicated. The discharge from the turbines under this condition increases but slightly and is indicated by the vertical projection of the point of greatest head (*m*) on the turbine discharge line (at *n*) which is so slightly above the 7,000 cubic feet line as to be hardly distinguishable on the diagram.

The power of the plant depends upon the head and the discharge through the wheels, hence the theoretical power which might be developed by the twenty-four turbines with a flow of 8,700 cubic feet per second would be about 13,800 H. P., which can be determined by calculation or is shown by the relation of the point *n* to the power curves. The actual value of these various points is more clearly shown on the second scale to the right, marked "Theoretical Power of Plant 24–57″ Turbines." A further increase in the discharge decreases the head until for the twenty-four turbines a minimum is reached at a discharge of 42,500 cubic feet per second. Under this condition of head the discharge through the wheels has also been somewhat reduced, and the corresponding horse power is reduced to 11,300 as shown by the intersection of the discharge curve and the line indicating the head existing under these conditions.

99. **Effects of Low Water Flow.**—In the case of low water when the flow is not sufficient to maintain the flow over the dam, if the turbines are run at full capacity, the water level behind the dam will drop until a point of equilibrium is attained where the head is just sufficient to force the entire discharge through the turbines. As the water level is lowered below the crest, the power of the plant rapidly diminishes owing to the great decrease in the head for a small decrease in the flow. When the head decreases beyond a certain point the power of the plant may be increased by closing some of the gates of the turbines until the discharge through the turbines is less than the discharge of the river, when the head will increase by the backing up of the water behind the dam.

Thus it will be seen by the diagram that, with only 6,000 cubic feet per second flowing in the river, if all of the turbines are operated the head will drop to about 12.7 feet, and the power of the plant under this head and flow would be about 8,660 H. P. If, under these conditions, one unit of six turbines, amounting to one-fourth of the plant, is shut down, the water will rise until the head is increased to about eighteen feet. Under these conditions about 800 cubic feet per second of this water will waste over the dam, and the power developed by the remaining portion of the plant will be 10,630 H. P., or, about 2,000 H. P. more with one unit shut down and with the resulting head than with all units in operation and the consequent lower head. The above discussion simply illustrates the point that it is rarely desirable to draw down the head of an operating plant, at least to any great extent, for the sake of operating a greater number of wheels, unless this is done for the purpose of impounding the night flow for use during the day or at times of maximum load. Even in this case too great a reduction in the head is undesirable and uneconomical.

100. Effects of Number of Wheels on Head and Power.—A diagram may be constructed which will show how the head on the wheel may be maintained by shutting off some of the wheels in case the flow becomes so small as to entirely pass the wheels, and thus reduce the head as previously described. An example of such a diagram is shown in Fig. 110, page 200, which is that portion of Fig. 109 enclosed by the dashed lines, enlarged and with additional lines which show the discharge and head curves for various numbers of wheels from four to forty-eight and with an additional number of horse power curves. These latter are not the same curves as are shown in Fig. 109, since they are computed on the basis of a plant efficiency of seventy-five per cent. of the theoretical power instead of simply the theoretical power as shown in Fig. 109. Other curves shown, namely "Height of Crest above Tail Water" and "Flow over Dam—All Gates Closed" are the same as shown in Fig. 109.

Two other lines on the diagram, namely curve *A* and curve *B*, show the limiting conditions of economic operation, and their location and construction will now be explained.

The problem to be solved in locating points on the lines is to find that point at which the same river discharge produces the same amount of power in a group of turbines one unit less in number than the group considered. This may be solved graphically in the following manner:

First, draw horizontal river discharge lines quite close together, say

at each 200 second feet, in the diagram considered; from the intersection of these lines with the "Turbine Head Curves" to the right of the "Height of Crest" line, drop perpendiculars to intersect the "Turbine Discharge Curve." Of these latter, find the one which lies on the same power line as does the intersection of the corresponding river discharge line with the turbine discharge line of the next higher num-

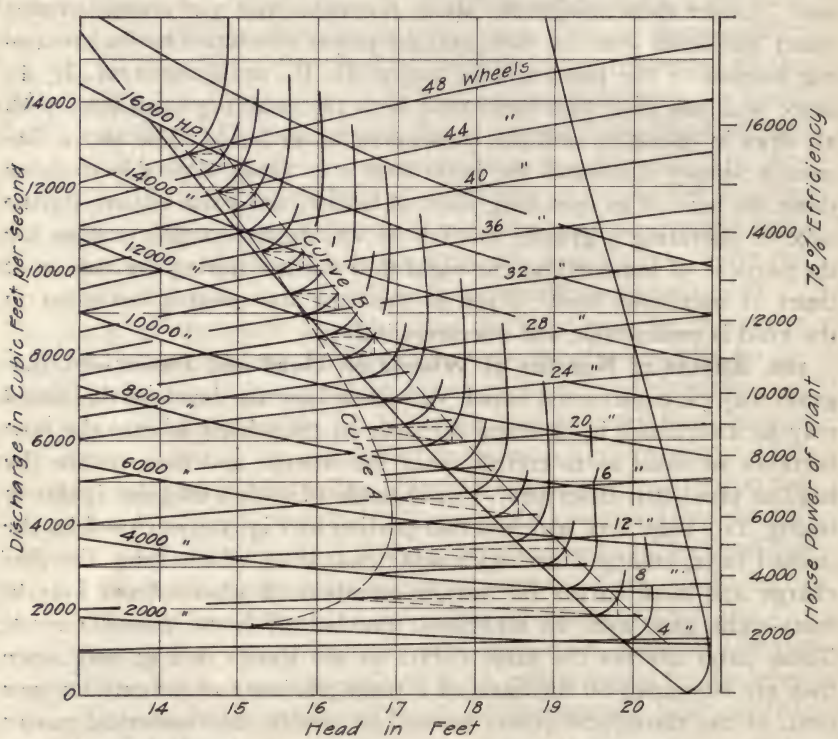


Fig. 110.—Relation of Number of Wheels to Power and Head (see page 199).

ber of turbines. The points which will fulfill these measurements will ordinarily lie between the lines drawn as described, but the lines drawn horizontally and vertically will assist in determining the location of the points.

This construction results in forming closely a right triangle, departing from the true triangle because of the hyperbolic form of the power curve which forms the hypothenuse. The right angle of the triangle is located on the head curve for a total number of machines at a point where this number of wheels will give the same power at the same rate

of river discharge as will a group of wheels containing one more unit. The smallest angle of the right triangle is located on the discharge curve for a certain total number of turbines which will produce the same power at the same rate of river discharge as will a group of turbines one unit less in number.

A series of such points determine two lines on the diagram which indicate the economic limits of operation of the plant: that is to say, these lines show the conditions at which the plant should be operated in order to produce the greatest amount of power. For example, assume a river flow of 10,000 second feet. It is at once seen that this will operate thirty-six machines to the best advantage, since if

34 turbines produce....................	13,100 horse power
36 turbines produce....................	13,200 horse power
38 turbines produce....................	12,000 horse power

Thirty-six turbines being the proper number to operate under the assumed conditions, there is very little flow over the dam since the turbines take about 9,950 second feet at the head of 15.55 feet indicated by the point of intersection if the 10,000 second foot river discharge line with the head curve for thirty-six turbines. Now should the river discharge decrease somewhat so that the power output of the thirty-six machines is about 13,100 H. P., it is seen that the same amount of power can be produced by operating thirty-four turbines. This number will require less water than the river flow, and in consequence the head will rise to slightly more than sixteen feet, and about 450 second feet will pass over the dam. Similarly, if the river discharge decreases to about 9,450 second feet, while thirty-four wheels are in operation, giving 12,600 H. P., the same power can be produced by closing down two wheels and operating thirty-two wheels under the head which will result from this rate of river discharge, namely about 16.3 feet.

A study of the diagram will show that if the flow decreases and the head on a certain number of turbines is allowed to fall until the point representing the conditions of operations lies to the left, or below the limiting line, curve *A*, a greater amount of power can be produced by closing down some of the wheels and operating a less number at an increase in head.

On the other hand, if the point representing the operating conditions for a certain number of wheels falls to the right or above the upper limiting line, curve *B*, the power output will be increased by increasing the number of wheels acting. For example, consider that twenty-four wheels are operating with a river discharge of 7,600 second feet. The

power production will be about 10,300 H. P. at a head of 17.25 feet. Twenty-six wheels operating at the same river discharge will produce 10,800 H. P. at a head of 16.8 feet, according to the diagram.

Thus it is seen that in order to operate the plant to produce the maximum amount of power, the operation should be confined to conditions as shown by a diagram of this sort between the two limiting lines marked as curve *A* and curve *B*.

In order to secure more accurate results a small correction for the variations in efficiency under the variations in head may sometimes be desirable. In the problem under consideration this is unnecessary on account of the small variation which takes place. However, when the

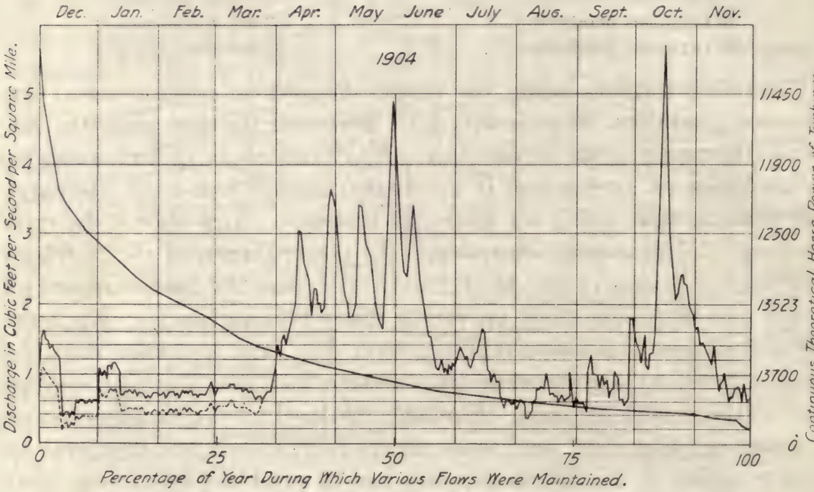


Fig. 111.—Power Hydrograph of the Wisconsin River at Kilbourn, Wis.

variations in head are considerable, this correction is essential if a close estimate of power at different heads is desired.

101. The Hydrograph Showing Variations in Power With Both Flow and Head.—Fig. 111 shows a power hydrograph similar to those previously discussed but with such changes in the scale as are necessary to show the continuous power that could have been developed by the four sets of turbines installed at Kilbourn, Wisconsin, during the year 1904, under the variations in head which would actually have occurred with a 300 foot dam with a fixed crest.

From the previous discussion of the conditions at Kilbourn it is seen that with a dam with fixed crest the variations in head, due to variations

in flow, are comparatively small (see Fig. 105, page 189 and Table 20, page 190). Consequently the power of the wheels would not decrease with an increase in flow to as great an extent as usually occurs in water power plants. With a system of flash boards or an adjustable crest to prevent overflow of lands above the dam at times of flood, the power of the plant when the crest is lowered would be still further reduced.

The curve showing the change of head with the change in flow

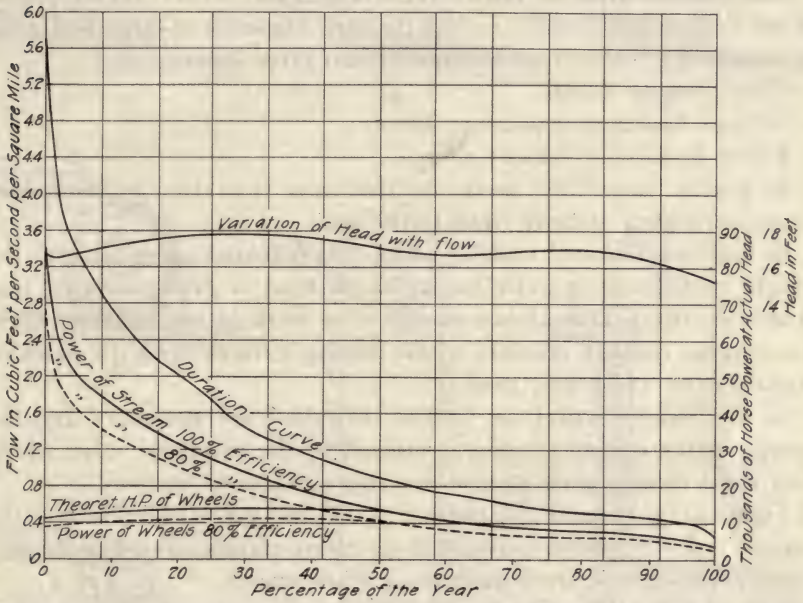


Fig. 112.—Variation in Power and Head With Changes in Flow as Applied to Kilbourn Conditions.

(Fig. 106, page 191) may be combined with the duration curve previously described (see Fig. 77, page 142) and a theoretical power curve derived from their relations. For an actual power curve, the decrease of head due to the passage of the water through the turbine, instead of over the crest of the dam (see Fig. 110, page 200), and the decrease in power due to the limited flow utilized and to the loss of energy in the wheels must be considered. These relations are illustrated, for Kilbourn conditions and for the year 1904, in Fig. 112.

CHAPTER IX

WATER WHEELS

102. Classification of Water Wheels.—Water wheels include most of the important hydraulic motors that are adaptable to large hydraulic developments. They may be divided into three classes, viz:

First: Gravity wheels.

Second: Reaction or pressure wheels.

Third: Impulse or velocity wheels.

In gravity wheels the energy of the water is exerted by its weight acting through a distance equal to the head.

In both reaction and impulse wheels the potential energy due to the weight of the water under the available head is first converted into kinetic energy. This kinetic energy does work in the reaction wheel through the reactive pressure of the issuing streams upon the movable buckets from which they issue.

In the impulse wheel the nozzles or guides are stationary and the energy of the issuing streams is utilized by the impulsive force which they exert in impinging against movable surfaces or buckets.

Figures 113, 114 and 115, pages 205, 206 and 208, which illustrate the various types of wheels included in the above classes, are adapted, with many modifications, from Reuleaux's "Constructor." *

103. Gravity Wheels.—Fig. 113 shows the various types of gravity water wheels or those wheels that are driven by the weight of the water. At moderate velocity, these motors are practically operated by gravity only, although under some conditions the impulse due to the velocity of the entering water may have an appreciable effect. In Fig. 113, A is an undershot water wheel; B is a half-breast wheel (see also Figs. 3 and 4, pages 3 and 4), and C is a high breast wheel. D is an overshot wheel. In C and D the buckets should be so designed as to retain the water until they reach the lowest point in the revolution of the wheel. E in this figure illustrates Duppinger's side-fed wheel. F illustrates an endless chain of buckets which is essentially the same in principle as the overshot wheel. G is a similar arrangement using

* "The Constructor." F. Reuleaux—trans. by H. H. Suplee, Philadelphia, Pa., 1893.

discs running with as small a clearance as possible in a vertical tube. When the water acts only by gravity, the wheels represented by A to E, inclusive, are only practicable when the wheel can be made as large or larger in diameter than the fall of the water. Where small diameters must be used, the arrangements shown in F and G are available.

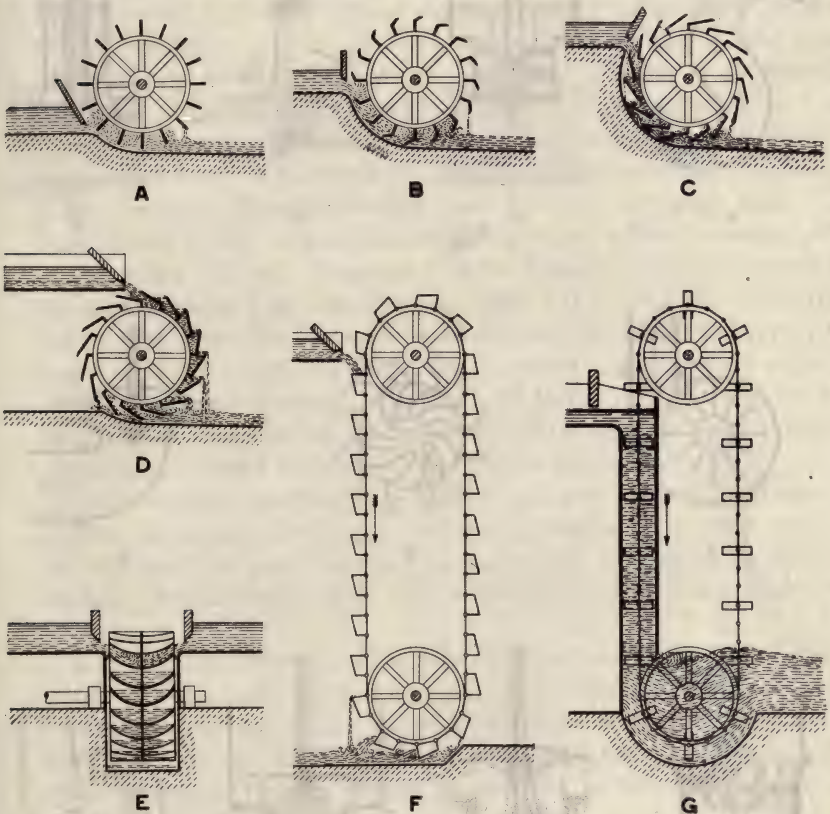


Fig. 113.—Diagrams of Gravity Wheels.

Very small wheels acting under high pressures may be employed by making use of the so-called chamber wheels, illustrated in H, I and J.

104. **Reaction Wheels.**—The wheels illustrated by the diagrams in Fig. 114 are of the second class or reaction wheels. Diagram A illustrates Barker's Mill of the form known as the Scotch turbine illustrated also by Fig. 8 (see page 7). This form of turbine is known in Germany as the Segner wheel. The water enters the vertical axis and discharges through the curve arms. B represents a screw turbine

which is entirely filled with water. C shows a Girard current turbine which has a horizontal axis and is only partially submerged. D is Cadiat's turbine with central delivery. It resembles the Fourneyron

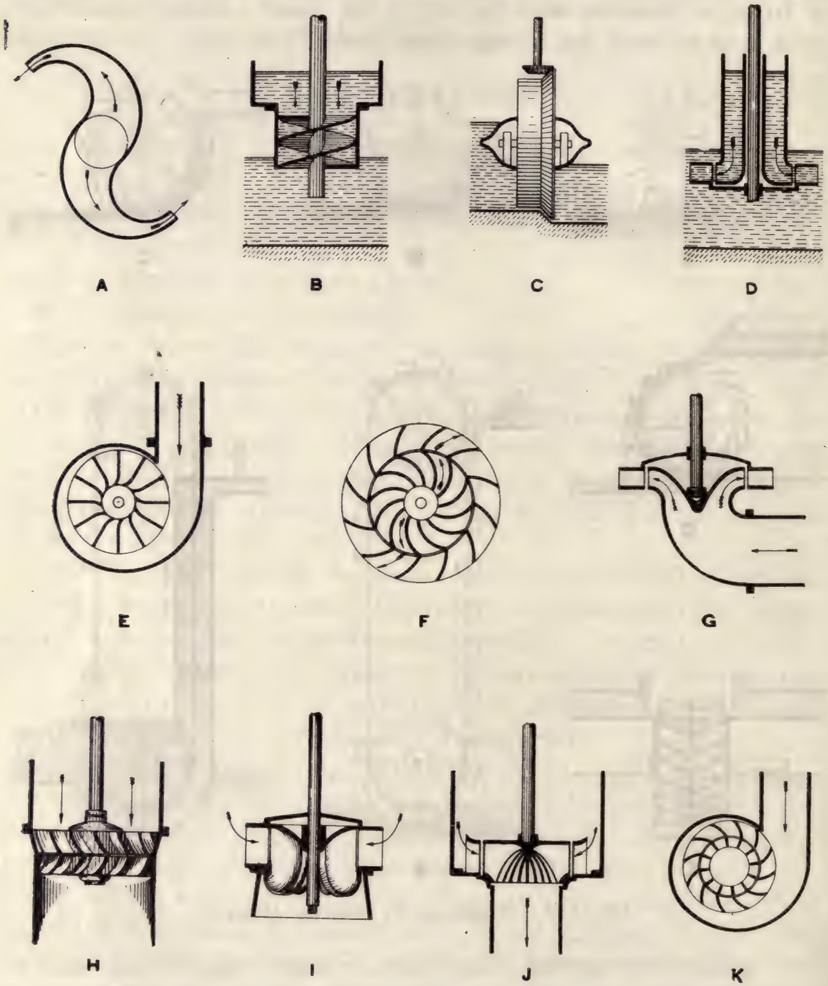


Fig. 114.—Diagrams of Reaction Wheels.

turbine except that there are no guides to direct the flow into the buckets. E is Thompson's turbine with circumferential delivery and horizontal axis. The discharge from this turbine is about the axis at both sides.

In diagrams A, B, C, D and E the column of water is received as a whole and enters the wheel undivided. The remainder of the forms illustrated in Fig. 114 show wheels in which the flow is divided into a number of separate streams by guides interposed in the streams before the water enters the wheel. Diagram F illustrates the Fourneyron turbine which acts with central delivery. The guide vanes are fixed and the discharge of the water is at the circumference of the wheel. The ordinary vertical form of the Fourneyron turbine is illustrated in Fig. 114. Diagram G, also in Fig. 114, is a modification of the Fourneyron turbine in which the water is being delivered upward from below. This form is sometimes called the Nagel turbine. Diagram H is the Jonval or Henschel turbine (see also Fig. 121, page 220). The guide vanes in this turbine are above the wheel which is entirely filled by the water column. Diagram J is the Francis turbine in practically its original form (see also Fig. 14, page 12). Diagram I illustrates the present American form or modification of the original Francis turbine. K is the Schiele turbine, a double wheel with circumferential delivery and axially directed discharge. In forms H, I, J and K, a draft tube may be used below the wheel to utilize any portion of the fall which occurs below the level of the bottom of the wheel.

In all reaction turbines, the water acts simultaneously through a number of passages around the entire circumference of the wheel. In the impulse or velocity turbine, the water may be applied to all of the buckets simultaneously or to only a portion of the circumference at a time.

105. Impulse Wheels.—The wheels illustrated in Fig. 115 are the third class of wheels which are driven by the impulse due to the weight of water acting through its velocity. Of these wheels, A is the current wheel or common paddle wheel. The paddles are straight and either radial or slightly inclined toward the current, as in the illustration (see also Figs. 1 and 2, page 2).

Diagram B is Poncelet's wheel (see also Fig. 5, page 4). The buckets run in a grooved channel and are so curved that the water drives upward and then falls downward, thus giving a better contact.

Diagram C shows an externally driven tangent wheel. The buckets are similar to the Poncelet wheel but with a sharper curve inward. The discharge of the water is inward. D is an internally driven tangent wheel similar to the preceding but with an outward discharge.

E is the so-called hurdy-gurdy or tangential wheel. The water is delivered through a nozzle and the wheel is practically an externally

driven tangent wheel of larger diameter and with a smaller number of buckets.

Diagrams F, G and H illustrate three types of impulse wheels with inclined delivery (see also Figs. 6, 7, 9 and 10, pages 5 to 9). Dia-

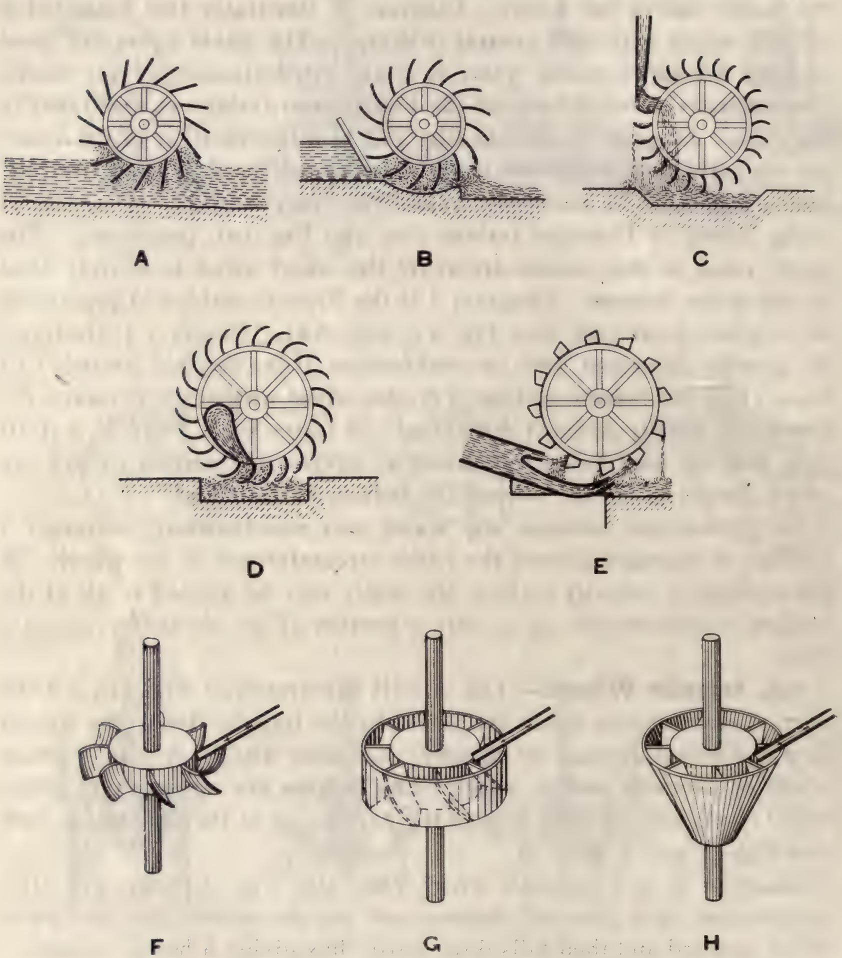


Fig. 115.—Diagrams of Impulse Wheels.

gram F shows a crude form of vertical wheel similar in form to the Indian wheel, Fig. 6. It is used on rapid mountain streams and is probably the original conception from which the turbine has been developed. Diagram G is the Borda turbine and consists of a series of spiral buckets in a barrel-shaped vessel. Diagram H is a Danaide

turbine which has spiral buckets enclosed in a conical tube. This is an old form of wheel formerly used in France.

106. Use of Water Wheels.—Almost all water wheels in practical use are modifications of some of the above forms and by a study of these forms a wheel may be classified and a clearer understanding obtained of the principles of its operation. Many of the forms of wheels shown in Figs. 113, 114 and 115 are practically obsolete or are used only in minor plants or for special conditions that make them of only general interest in the study of water power.

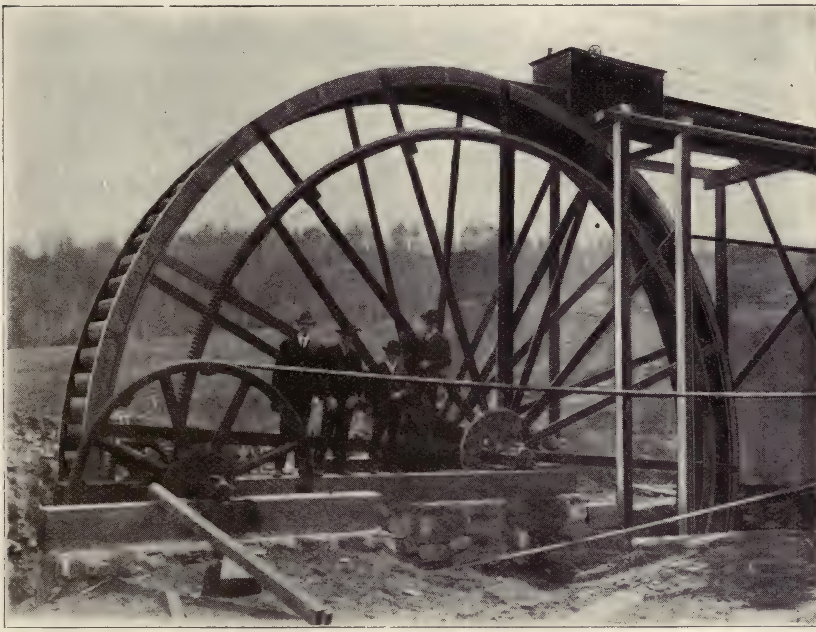


Fig. 116.—"Overshot" Water Wheel. Manufactured by Fitz Water Wheel Co.

While gravity wheels are still occasionally used their application is entirely to the smaller water power plants. In many cases the turbines purchased for such installations are of cheaper make, poorly designed, constructed and selected, and often improperly set and, consequently, inefficient. In such cases, and where the question of back water and the interference of ice is not important, the gravity wheel may be more efficient and quite satisfactory. Well designed and well constructed gravity wheels are said to give efficiencies of eighty-five per cent. and above (see Fig. 18, page 20, and Fig. 116). With such plants the en-

gineer has usually little to do and consequently they will not be further considered here. The types of wheels now most largely used for moderate and large water power developments are the reaction and impulse turbines.

107. Classification of Turbines.—All modern turbines consist of a wheel to which buckets are attached and which is arranged to revolve in a fixed case having attached to it a nozzle, guide or series of guides. The guide passages or nozzles direct the water at a suitable angle onto the buckets of the wheel. The revolving wheel contains curved buckets or passages whose functions are to receive the water, utilize its energy and discharge or waste it as nearly devoid of energy as possible.

Turbines may be classified in various ways:

First: In accordance with the action of the water on the same.

(A) *Reaction or pressure turbines,* such as the Fourneyron, Jonval, Francis, etc. (see Fig. 114, page 206, G, H, I and J).

(B) *Impulse or velocity turbines,* such as the Girard and tangential wheels (see Fig. 115, page 208, diagrams D and E).

(C) *Limit turbines,* which may act either by reaction or impulse.

Second: In accordance with the direction of flow in reference to the wheel.

(A) *Radial flow turbines.* In these turbines the water flows through the wheel in a radial direction. These may be subdivided into—

(a) *Outward radial flow turbines,* such as the Fourneyron and Cadiat (see Fig. 114, diagrams F and D).

(b) *Inward radial flow turbines,* or wheels in which the water flows inward in a radial direction such as the Francis and Schiele turbines (see Fig. 114, J and K).

(B) *Axial flow turbines* in which the general direction of the water is parallel to the axis of the wheel such as the Jonval and Girard wheels of similar design (see Fig. 114, H).

(C) *Mixed flow turbines,* or turbines in which the flow is partially radial and partially axial as in turbines of the American type (see Fig. 114, diagram I; also Figs. 125 to 142 inclusive).

Third: In accordance with the position of the wheel shaft.

(A) *Vertical* (see Figs. 118, 120, 121, 133, etc., pages 217 to 226).

(B) *Horizontal* (see Fig. 134, page 227).

Fourth: In accordance with the arrangement of nozzles or guides.

(A) *Complete turbines* with guides surrounding the entire wheel.

(B) *Partial turbines* with guides partially surrounding the wheel in one or more groups.

The reaction turbine is a turbine with restricted discharge which acts through the reactive pressure of the water. Under some conditions the energy of the water may be exerted, at least in part, by virtue of its velocity. The impulse turbine receives energy practically entirely through the reactive pressure on the buckets caused by changing the direction of flow of the moving mass of water, which moves over the surface of the bucket with the velocity due to the head, since the discharge is unrestricted. The limit turbine may act entirely as a reaction or as an impulse turbine according to the conditions under which it operates.

108. Condition of Operation.—These wheels operate under the following conditions:

REACTION OR PRESSURE TURBINES.

Guides complete.

Buckets with restricted outlets.

Buckets or wheel passages completely filled.

Energy most largely developed through reactive pressure.

Discharge usually below tail water or into a draft tube.

IMPULSE OR VELOCITY TURBINES.

Guides partial or complete.

Buckets with outlets free and unrestricted.

Wheel passage never filled.

Energy entirely due to velocity.

Discharge must be above tail water.

No draft tube possible, except with special arrangement which will prevent contact of tail water with wheels.

LIMIT TURBINES.

(A) Buckets so designed that the discharge is unrestricted when above tail water.

Buckets in this case are just filled. Act primarily by effect of velocity.

Discharge above tail water.

(B) If tail water rises to buckets, the discharge is restricted and pressure reaction results.

In this case the full bucket admits reaction and discharge may be below tail water.

109. Relative Advantage of Reaction and Impulse Turbines.—The reaction wheel is better adapted for low and moderate heads, especially

when the height of the tail water varies and where the amplitude of such variation is a considerable percentage of the total head. Such a wheel, which is designed to operate with the buckets filled, can be set low enough to utilize the entire head at all times and will operate efficiently when fully submerged. The reaction wheel can therefore be set to utilize the full head at time of low tail water and when the quantity of flow is limited. For low head developments this is an important factor. The impulse turbine, on the other hand, must have a free discharge and must therefore be set far enough above the tail water to be free from back water if it is to be operated at such times.

Another difference between the reaction and the impulse turbine is the higher relative speed with which the former operates. This is often a distinct advantage, for direct connection with high speed machinery, and with low and moderate heads. On the other hand, with high heads the slower speed of the impulse wheels is frequently of great advantage, especially in the form of the tangential wheel when the diameter can be greatly increased and very high heads utilized with moderate revolutions. In such cases the height of the back water is usually but a small percentage of the total head, and the loss due to the higher position of the wheel is comparatively small.

The speed of a wheel for efficient service is a function of the ratio of the peripheral velocity of the wheel to the spouting velocity of water under the working head. This ratio will ordinarily vary from .60 to .95 in reaction turbines, according to the design of the wheel. In impulse turbines this ratio varies from .43 to .48.

110. **Relative Turbine Efficiencies.**—The impulse turbine has the further advantage of greater efficiency under part gate,—that is, at less than its full capacity. When, as is usually the case, a wheel must operate under a variable load it becomes necessary to reduce the discharge of the wheel in order to maintain a constant speed with the reduced power required (see Fig. 117, page 213). This is accomplished by a reduction in the gate opening which commonly greatly affects the economy of operation.

The comparative efficiencies of various types of the turbines are shown in Fig. 117. The maximum efficiency of turbines when operated at the most satisfactory speed and gate will be about the same for every type, if the wheel is properly designed and constructed and the conditions of operation are suitable for the type used. This maximum efficiency may vary from seventy-five to eighty-five per cent., or even between wider limits, but, with suitable conditions, should not be

less than eighty per cent. In order to make the curves on the diagram truly comparative, the percentage of maximum efficiency and of maximum discharge are plotted instead of the actual efficiencies and actual discharge.

The Fourneyron turbine usually shows very poor efficiencies at part gate as shown in Fig. 117. The curve for this turbine is drawn from

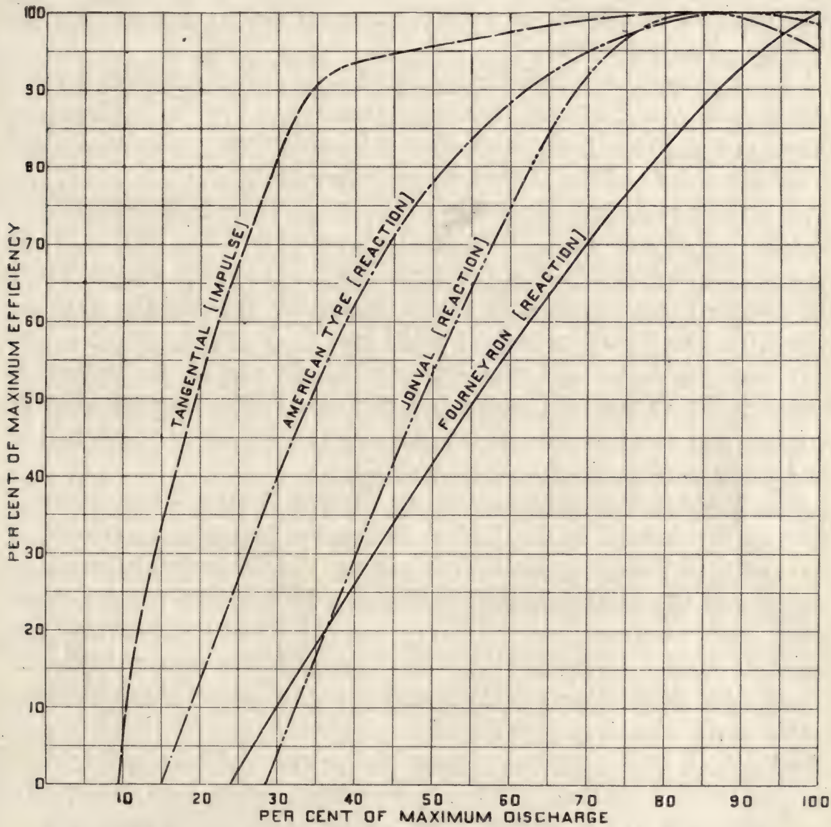


Fig. 117.—Comparative Efficiencies of Various Types of Turbines.

Francis' test of the Tremont (Fourneyron) turbine (see Fig. 118, also Appendix B, page 744) and is substantiated by efficiency curves shown by various tests by James Emerson.*

The Jonval turbines usually show better part gate efficiencies than the Fourneyron but are not as efficient, under such conditions, as turbines of the inward flow or Francis type. The Jonval curve, shown

* See "Hydrodynamics" by James Emerson.

in Fig. 117, is plotted from the test made in 1884 at the Holyoke testing flume * of a thirty-inch regular Chase-Jonval turbine (see Appendix B, page 746).

The American-Francis turbine varies greatly in part gate efficiency according to the details of design and the relation of speed and head under which it operates. The curve shown in Fig. 117, representing this type, is from the test of a wheel manufactured by J. & W. Jolly of Holyoke, Massachusetts.

The impulse wheels when properly designed and operated show a higher part gate efficiency than any other type of wheel. The curve shown in Fig. 117 is from a test of a twelve-inch Doble tangential wheel in the laboratory of the University of Wisconsin.†

As already indicated, the design of the wheel has a great influence on its efficiency at part gate. Individual wheels or series of wheels of any type may therefore depart widely from the curves above shown, which are intended only to show as fairly as possible the usual results obtained from well made wheels of each type.

It should be noted also that efficiency is only one of the factors influencing the choice of a wheel and that many other factors must be weighed and carefully considered before a type of wheel is selected as the best for any particular set of conditions.

111. **Turbine Development in the United States.**—The development of the turbine in the United States has been the outgrowth of some seventy years of practical experience. In the early settlement of the country the great hydraulic resources afforded facilities for cheap power and numerous water powers were developed under low and moderate heads. These developments created a correspondingly great demand for water wheels and stimulated invention and manufacturing in this line. American inventors have devised many different forms of wheels which were patented, constructed, tested and improved to meet the prevailing conditions. When a successful wheel was designed, it was duplicated in its original form and its proportions increased or diminished, to conform to the desired capacity. As wheels of greater capacity or of higher speed have been required, modifications have been made and improved systems have resulted.

The best American water wheel construction began with the Boyden-

* See page 44 of 1897 catalogue of Chase Turbine Manufacturing Co., Orange, Mass.

† From "Test of a 12″ Doble Tangential Water Wheel," an unpublished thesis by H. J. Hunt and F. M. Johnson.

Fourneyron and Geylin-Jonval turbines of improved French design, but modern American practice began to assume its characteristic development with the construction of the Howd-Francis turbines, already described. Moderate changes in the form and arrangement of buckets and other details gave rise to the earlier forms of "Swain," "Leffel" and "American" wheels each of which consisted of an inward flow turbine modified from the earlier designs of Howd and of Francis as the experience of the inventor seemed to warrant. In all of these cases the wheels discharged inward and essentially in a radial direction and had to be built of sufficient diameter to provide an ample space for receiving the discharging waters. This necessitated slow speed wheels of comparatively low capacity (see Table 1, page 14). In order to secure higher speed, the diameters of the wheels were reduced, thus reducing the power. This reduction was, however, more than counterbalanced, in the later wheels, by an increase in the width of the bucket in an axial direction. It was found also that the capacity of the wheels could be materially increased, with only small losses in efficiency, by decreasing the number of buckets. Wheels were gradually reduced in diameter and the buckets increased in breadth until, in many cases, they reached very nearly to the center of the wheel. This necessitated a downward discharge in the turbine and resulted in the prolongation of the buckets in an axial direction in many cases to almost double the width of the gate. From this development has resulted the construction of a series of wheels known as the "American turbines" having higher speed and greater power than has been reached in European practice.

The entire line of development has, until within the last fifteen years, been toward the increase of speed and power for low and moderate head conditions. It is only within this period that a considerable demand has been felt in this country for turbines having other characteristics and adapted for higher heads.

The American type of turbine, prior to 1890 was not designed or suitable for high heads, its origin being the result of entirely different conditions. About 1890 came a demand for turbine wheels under comparatively high heads which manufacturers of wheels of the American type were poorly equipped to meet. The first of such wheels supplied were therefore of European types, which apparently better suited such conditions. Recognizing, however, the importance of meeting such demands, the American manufacturer found that the wheels of essentially the original Francis type were well suited for this purpose. The nar-

row wheel and numerous buckets of the earlier types reduced the discharge of water, and, increasing the diameter, reduced the number of revolutions. Such types of wheels of high efficiency can now be obtained from the leading manufacturers in the United States, and, while some manufacturers still furnish only from their standard designs, which are suited only for the particular conditions for which they were designed, others are prepared to furnish special wheels which are designed and built for the particular conditions under which they are to be used.

The systems of wheels offered by American manufacturers, which can be readily and quickly duplicated from stock patterns at a much less expense than would result from the design of special wheels for each particular case, has enabled American manufacturers to furnish water wheels of a fairly satisfactory grade and at a cost which would have been possible in no other way. In the United States the cost of labor has been comparatively high and special work is particularly expensive, much more so than in Europe where skilled mechanics receive a compensation for labor which is but a small fraction of that of their American competitors. Average American practice, at the present time (1907) undoubtedly leaves much to be desired and considerable advance may be expected from the correction of designs, resulting from practical experience and by the application of scientific analysis.

During the last eight years (1907 to 1915), the improvement in turbine design and construction in the United States has been very great. The increase in the efficiency of turbines has been fully ten per cent., a maximum of 93.7 per cent. having been reached within the last year. High capacity has also been developed (see Tables 1 and 2, page 14) together with the ability to manufacture wheels of large diameter (see page 244 and Fig. 142, page 231), making possible the use of single vertical wheels where multiple vertical wheels were necessarily used ten years ago. At the present time (1915), the best American practice in turbine design and construction is fully abreast with the best European practice, and in some ways considerably in advance.

112. The American Fourneyron Turbine.—As noted in Chapter I, one of the first reaction turbines developed in the United States was the Boyden wheel of the Fourneyron type.

In these wheels (see Fig. 118) the water entered from the center, guided by fixed curve guides *g* (Fig. 119) and discharged outward through the buckets *B*. The use of these wheels gradually spread and they rapidly replaced many of the old overshot and breast wheels used

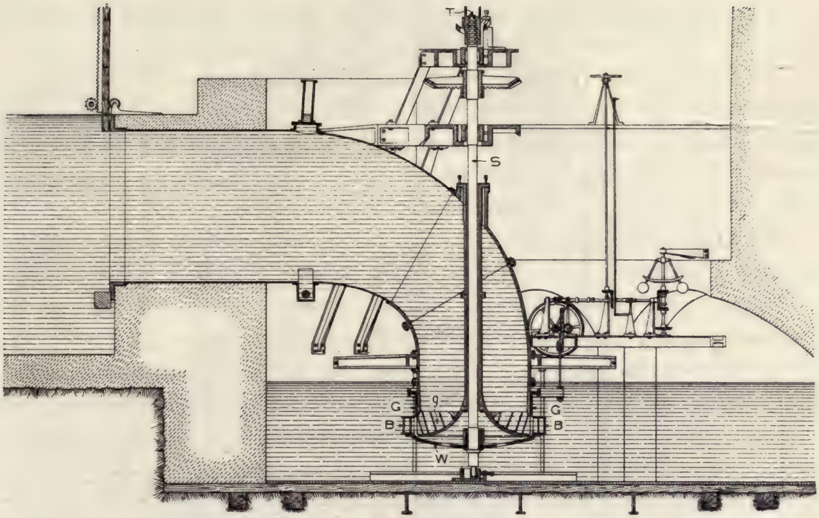


Fig. 118.—Tremont (Boyden-Fourneyron) Turbine. (After Francis.)

Fig. 119.—Guides and Buckets of Tremont (Boyden-Fourneyron) Turbine.

up to that time, and soon became the foremost wheel in New England.

The manufacture of the Fourneyron turbine has, for common use, been discontinued on account of the competition of other cheaper

wheels which were found to be more efficient at part gate and more generally satisfactory under ordinary conditions of service.

The Fourneyron turbine, when well designed and constructed, is a turbine of high full gate efficiency. This wheel is adapted for high heads where a comparatively slow speed is desired, and it is now frequently used for high grade and special work where its peculiarities seem best suited to such conditions.

One of the modern applications of the Fourneyron turbine is that in the power plant of The Niagara Falls Water Power Company. Fig. 120, page 219, shows vertical and horizontal sections of one of the double Fourneyron units used by this company in their first plant. These wheels discharge 430 cubic feet per second and make 250 revolutions per minute; at seventy-five per cent. efficiency each wheel will develop 5,000 H. P. The buckets of these wheels are divided vertically into three sections or stories in order to increase their part gate efficiencies. These wheels are of Swiss design by the firm of Faesch and Picard and were built by The I. P. Morris Company of Philadelphia. The wheels are vertical and connected by vertical shafts, each with one of the dynamos in the station above. The shaft is built of three-quarter inch steel, rolled into tubes thirty-eight inches in diameter. At intervals the shafts pass through journal bearings, or guides, at which points the shafts are reduced to eleven inches in diameter and are solid. The speed gates of these wheels are plain cylindrical rims which throttle the discharges on the outside of the wheels and which, with the cooperation of the governor, keeps the speed constant within two per cent. under ordinary conditions of operation. Another wheel of this type is that manufactured and installed at Trenton Falls, New York, by the same firm (see Fig. 322).

113. The American Jonval Turbine.—The Jonval turbine, originally of French design, was introduced into this country about 1850 and became one of the most important forms of turbine of early American manufacture. In the tests of turbines at Philadelphia in 1859–60 (see page —) a Jonval turbine developed the highest efficiency and the type was adopted by the city for use in the Fairmount Pumping Station. Like the Fourneyron turbine, these wheels, while highly efficient at full gate, have largely been superceded by other cheaper and more efficient part gate types, except for special conditions.

Figure 121 shows the Geylin-Jonval turbine as manufactured by the R. D. Wood Company of Philadelphia. *W* represents the runner, *B*

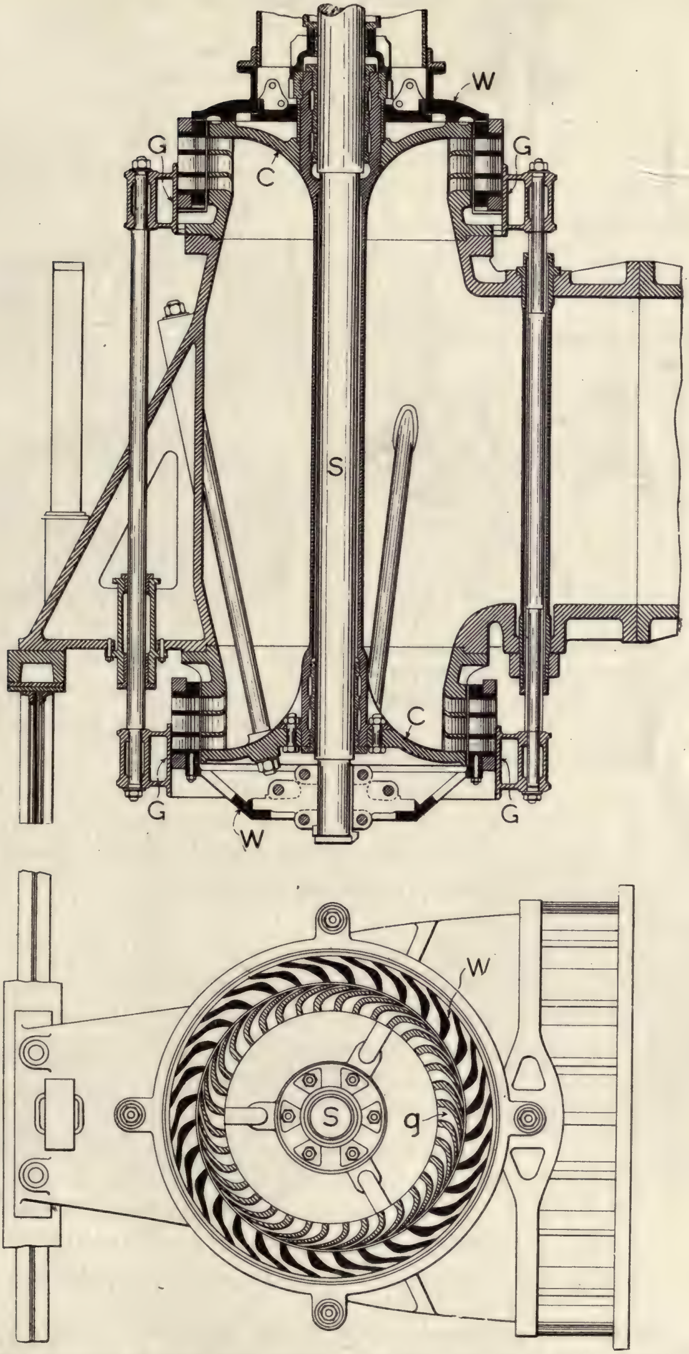


Fig. 120.—Double Fourneyron Turbine of The Niagara Falls Water Power Company. (Designed by Faesch & Picard; built by I. P. Morris & Co.)

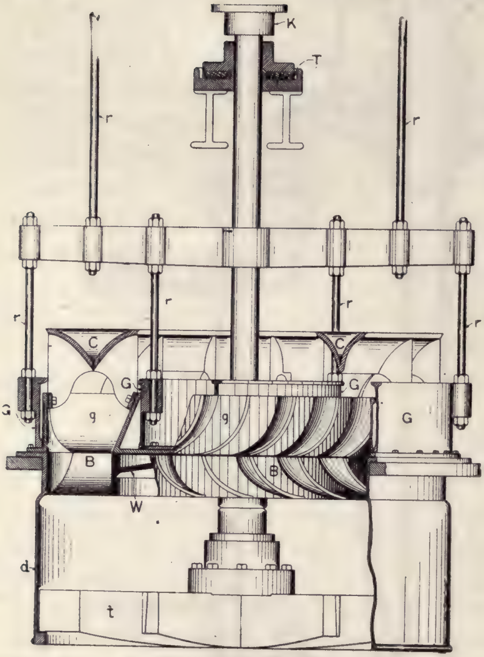


Fig. 121.—Vertical Geylin-Jonval Turbine (Manufactured by R. D. Wood & Co.)

the buckets which receive the water through the guides *g*. The wheel shown has double inlets that are closed by the double cylinder gates *GG*. This gate closes up against the hood *C*, by means of the rods *r, r*, which connect with the governor mechanism. The general design of the ordinary wheel of this type is perhaps best shown by Fig. 122, page 221.* In this figure *A* is the fixed or guide wheel and *B* is the movable or turbine runner.

In the later hydraulic developments the use of this wheel has been confined, largely at least, to locations that require special designs.

114. The American Type of Reaction Turbine.—The Howd Wheel (Fig. 13, page 11) from which the idea of the Francis inward flow wheel (Fig. 12, page 10) was derived, was invented in 1838 and acquired a considerable market throughout New England. From these wheels originated the American inward and downward or mixed flow turbines.

The early wheels of American manufacture were designed very much after the style of the Francis wheel with changes, more or less radical, in the shape and details of the buckets. The demand for wheels of greater power, and higher speed, has resulted in a gradual development of other and quite different forms.

The development of the turbine in the United States is well illustrated by that of the "American" turbine of Stout, Mills & Temple, now The Dayton Globe Iron Works Company. This wheel was designed in 1859 and was called the American Turbine. The general form of the original turbine wheel is shown in Fig. 123, page 221.

This was followed (1884) by the design of what is known as the

* See page 7, 1877 catalogue, J. L. & S. B. Dix, Glen Falls, N. Y.

"New American" turbine, illustrated by Fig. 124. In this wheel the buckets are lengthened downward and have a partially downward as well as inward discharge.

This wheel was followed in 1900 by the "Special New American" illustrated in Fig. 125, page 223, having a great increase in capacity and power.

The fourth and most recent type (1903) is the "Improved New American" illustrated in Fig. 126, page 223. The comparative power and speed of these various wheels is shown in the tables on page 222.

Table 22 is misleading to the extent that while the diameter of each wheel is given as forty-eight inches

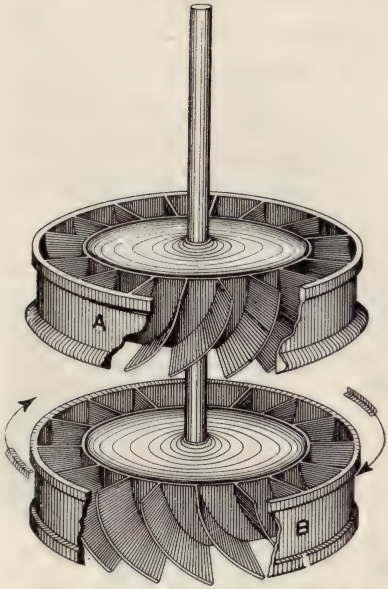


Fig. 122.—Jonval Turbine as Manufactured by J. L. & S. B. Dix.

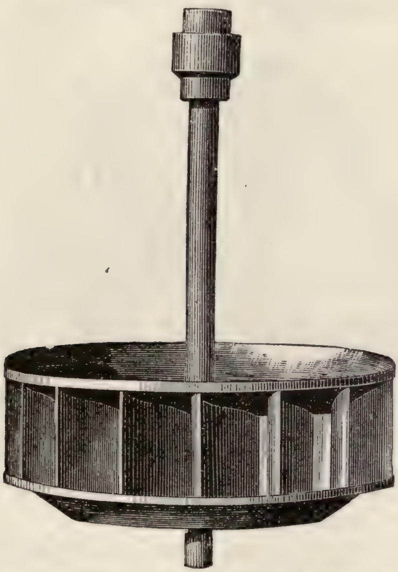


Fig. 123.—American Turbine Runner* (see page 220).

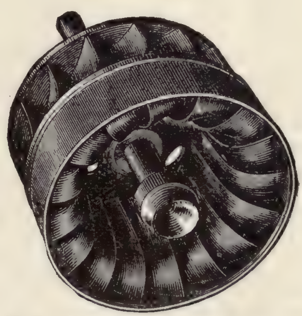


Fig. 124.—New American Turbine Runner.

* Manufactured by The Dayton Globe Iron Works Co.

such diameters are not strictly comparative. Part of the additional capacity and power of the "Special New American" and of the "Improved New American" is due to the cutting back of the buckets (see Figs. 123 to 126) which, while it reduces the diameter at the point of measurement, gives a discharge which would be fairly comparative with wheels of the older type of perhaps three or four inches larger diameters (see Section 120, page 246).

TABLE 22.

Development of "American" Turbines.—Capacity, Speed and Power of a 48-inch Turbine under a 16-foot Head.

	Year brought out.	Discharge in cu. ft.	Rev. per min.	Horse power.
American	1859	3271	102	79.1
Standard New American.....	1884	5864	102	141.8
New American...............	1894	9679	107	234.0
Special New American.......	1900	11061	107	267.0
Improved New American....	1903	13234	139	325.0

The development of turbines may also be illustrated by a comparison of the size and speed of turbines of various series required to develop essentially the same power (see Table 23).

TABLE 23.

Increase in Speed of "American" Turbines for Same Power (16-foot head).

	Size of wheel.	Horse power.	R. P. M.
American	48	79.1	102
New American..............................	36	81.5	136
Special New American.....................	$27\frac{1}{2}$	87.3	186
Improved New American..................	25	87.5	267

Figures 127 and 128, page 224. show a vertical and a horizontal half plan, half section of a vertical Improved New American turbine. *W* is the crown and hub of the wheel; *B*, the buckets; *G, G*, are the wicket gates that control the admission of water to the wheels and which are operated by means of the ring *Gr*, which is moved by an eccentric and rod *r*, connected with the governor through the shaft *P*.

The inner edges of the bucket are spaced some distance from the shaft and the main discharge is inward and downward, though a portion of the bucket will admit of a slightly outward discharge.

115. The Double Leffel Turbine.—Perhaps the greatest departure of American inventors from the lines of the original Francis type of turbine was that of James Leffel. In this wheel was combined a double runner, the upper half being a radial inflow runner of the Francis type and the lower half consisting of a runner with inward radial admission and axial discharge, essentially on the line of the later development of the American type of wheels. The wheel, as originally designed, had the narrow bucket, slow speed and low power of all early American wheels. In its later development the buckets have been extended inward and downward and these wheels have found their best modern development in the Samson-Leffel wheel, illustrated in Figs. 129 to 133, pages 225 to 226.

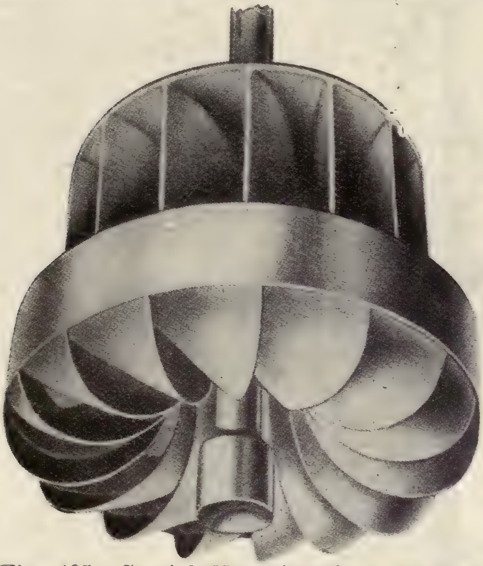


Fig. 125.—Special New American Turbine Runner * (see page 221).

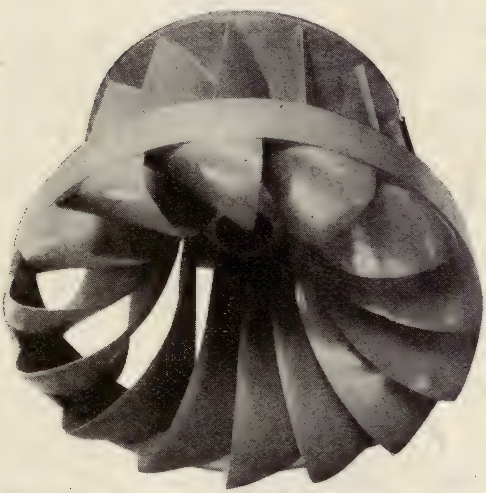
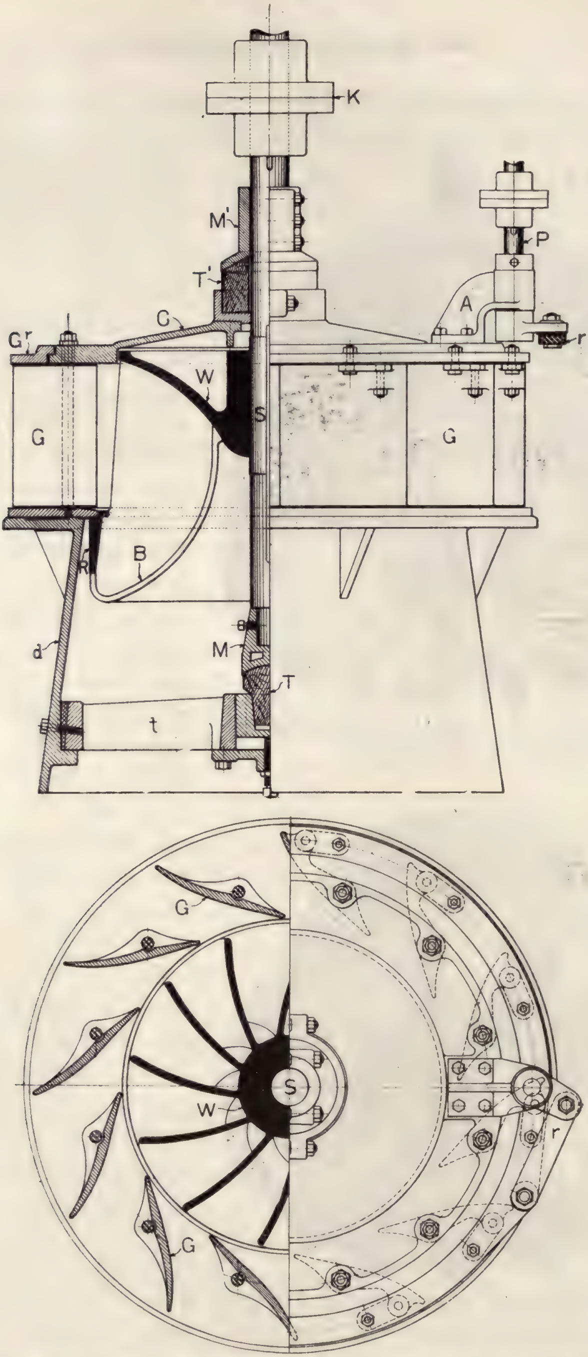


Fig. 126.—Improved New American Turbine Runner * (see page 221).

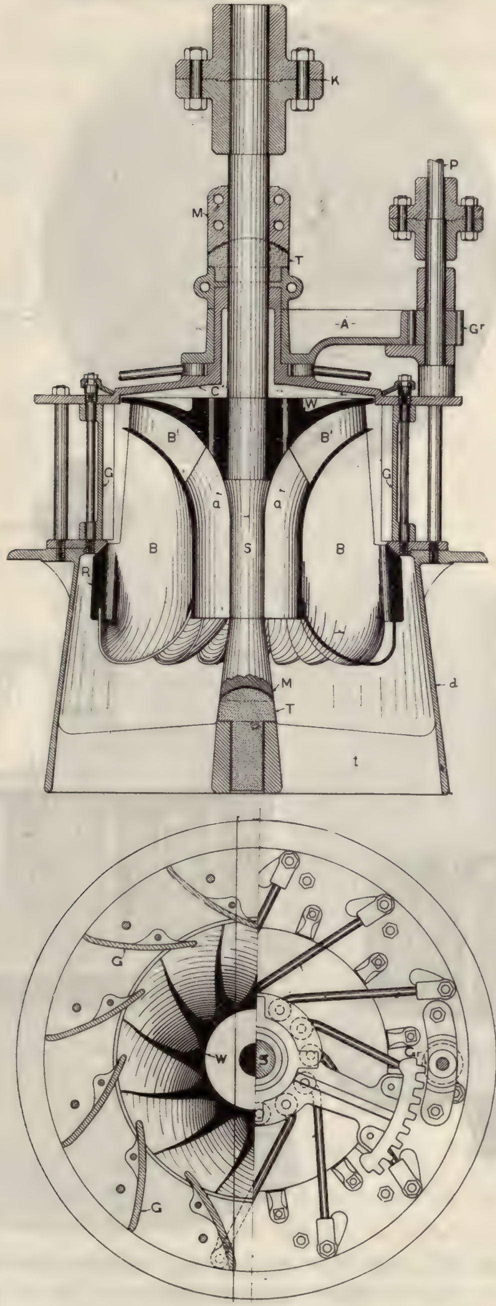
In Fig. 129, *W* represents the hub and crown of the wheel which is securely keyed to the shaft *S*. *B′B′* are the upper buckets that discharge inward and downward through the passage *aa*. The lower buckets *BB*, it will be noted have the same lines as other modern wheels of the American type. They receive the wa-

* Manufactured by The Dayton Globe Iron Works Co.



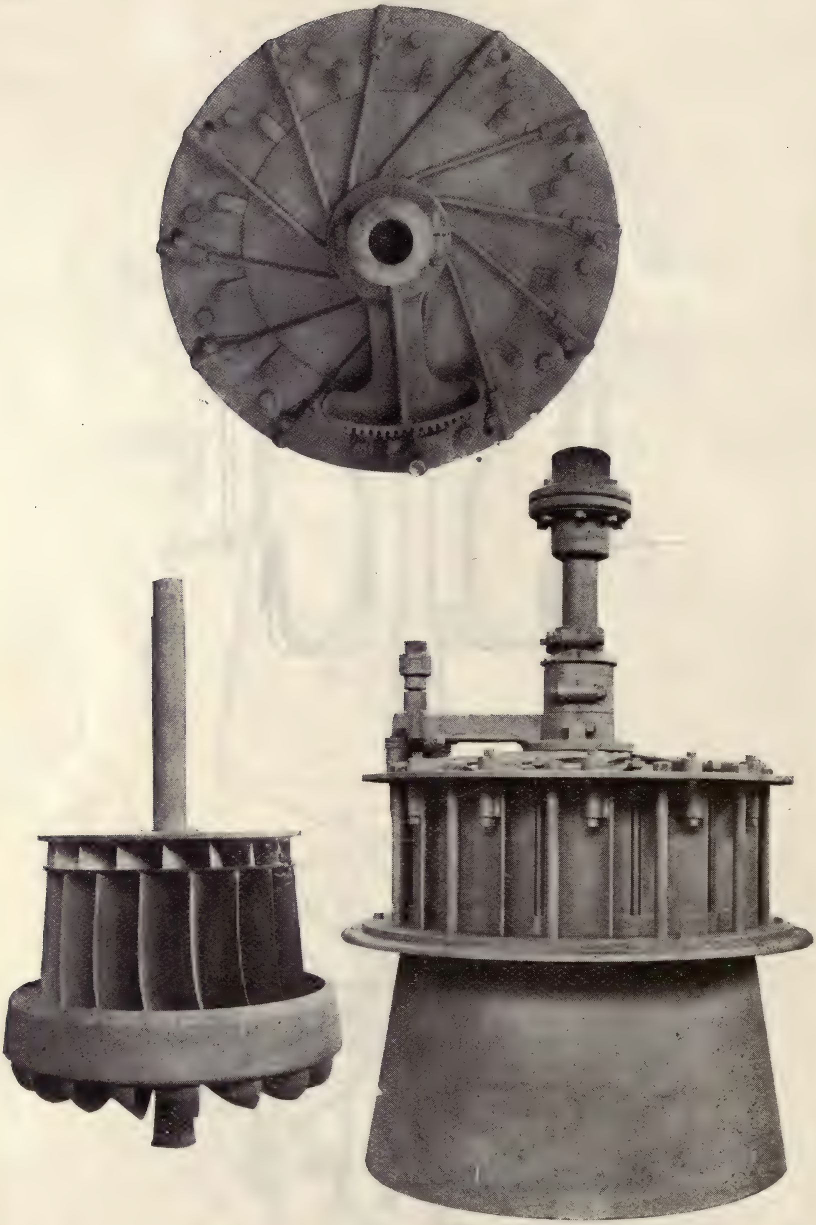
Figs. 127 and 128.—Section and Plan of Improved New American Turbine* (see page 222).

* Manufactured by The Dayton Globe Iron Works Co.



Figs. 129 and 130.—Section and Plan of Samson Turbine * (see page 223).

* Manufactured by The James Leffel & Co.



Figs. 131, 132 and 133.—Top View, Runner and Outside View of Samson Turbine * (see page 227).

* Manufactured by The James Leffel & Co.

ter inward and discharge it downward, outward and inward with the general purpose of distributing it over the cross-section of the turbine tube. The gates *G*, are of the wicket type and are conneccted by rods with an eccentric circle which is operated through the arm *A*, and the gearing *Gr*, by the governor shaft *P*. The gate gearing is well shown by reference to the section-plan, Fig. 130, and the top view, Fig. 131, page 226.

The Samson turbine runner as illustrated in Fig. 132 and Fig. 133 shows an outside view of one of the vertical turbine units. The de-

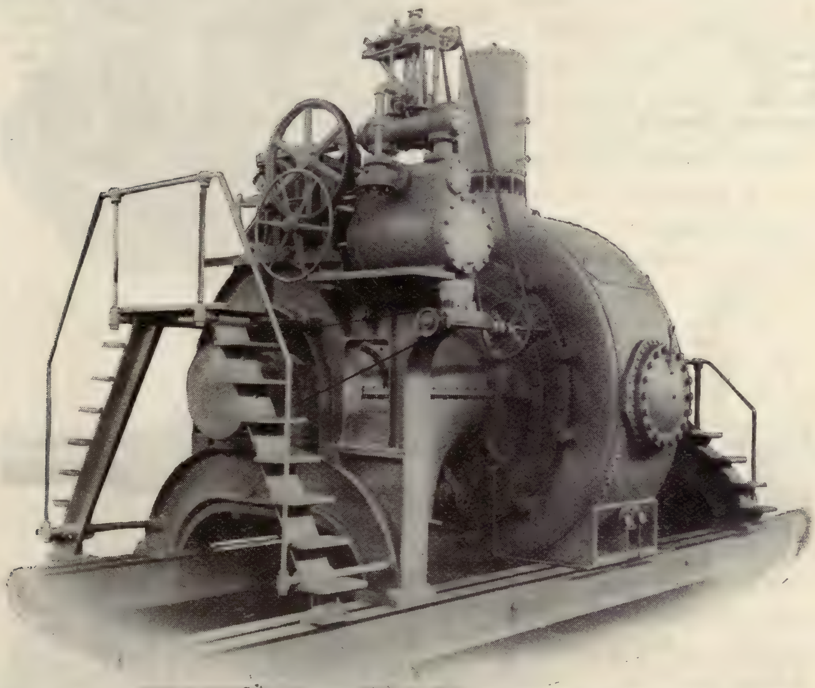


Fig. 134.—Double Horizontal Leffel Turbine of The Niagara Falls Hydraulic Power & Manufacturing Co. Manufactured by The James Leffel & Co. (see page 228).

velopment of this wheel is illustrated by Table 24. This table is fairly representative of the growth of this turbine as the diameter is, in all cases, the maximum diameter of the wheel (see Section 120, page 246).

The adaptability of the earlier turbine designs to the later moderate head developments is well illustrated in the design of the wheels for The Niagara Falls Hydraulic Power and Manufacturing Company,

installed by The James Leffel Company about 1892. These turbines have the single narrower buckets, smaller discharge and relatively slower speed of the earlier designs. The runners are double discharge, horizontal, seventy-four inches in diameter and operate at a speed of 250

TABLE 24.

Development of "Leffel" Wheel.—Capacity, Power and Speed of 40-inch Wheel Under 16-foot Head.

	Year brought out.	Discharge.	Rev. per minute.	Horse power.
Standard	1860	2547	138	64½
Special	1870	3672	138	93
Samson	1890	6551	158	155
Improved Samson...........	1897	8446	163	207

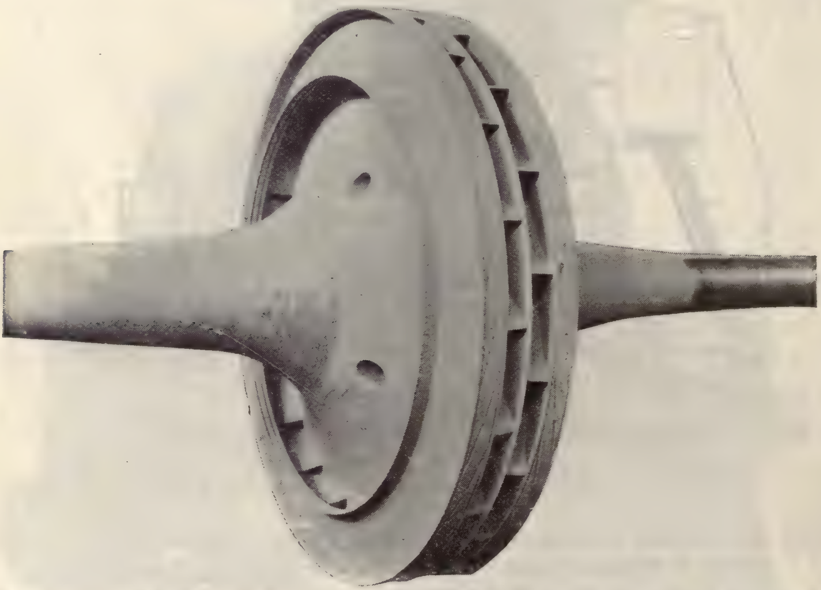


Fig. 135.—Leffel Double Runner of The Niagara Falls Hydraulic Power & Manufacturing Co. Manufactured by The James Leffel & Co. 74" 3100 H. P. 213 ft. head 250 R. P. M.

revolutions per minute under a head of 213 feet, and each wheel develops about 3,100 H. P.

Figure 134 shows one of these units complete. Fig. 135 is a view of the runner. For a test of this wheel, made December 1903, see Fig. 231.

116. Other American Wheels.—The development of modern American wheels could, perhaps, have been equally well illustrated by the growth of various other American turbines. The development of all American wheels up to the present time has been on the line of increasing both the speed and the power of the wheel for low head, with a return to the earlier type for wheels to be used under the moderate heads.

Fig. 136.—Hunt-McCormick Runner of The Rodney Hunt Machine Co.

Figure 136 illustrates a runner of the well-known McCormick pattern. Mr. J. B. McCormick, who had previously become familiar with certain wheels of large capacity designed and patented by Matthew and John Obenchain, re-designed and improved these wheels, about 1876, and secured high efficiencies together with increased power far beyond any other wheels of that period. McCormick wheels in their original or modified form are now made by a large number of American manufacturers and these wheels have had a marked effect on the design of almost all modern American water wheels. The runner in the illustration is the Hunt-McCormick runner as manufactured by The Rodney Hunt Machine Company, but is very similar to the McCormick wheels of various other manufacturers.

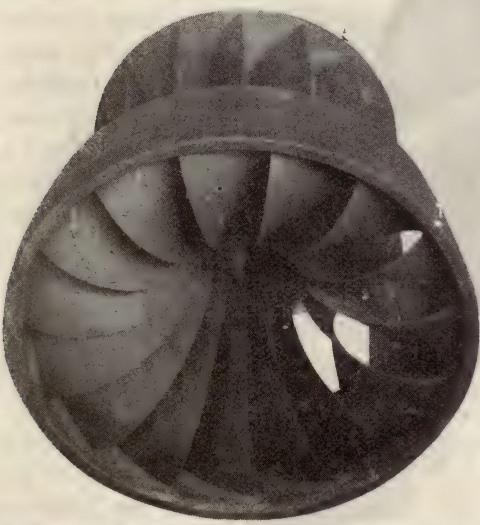


Fig. 137.—Smith Runner of S. Morgan Smith Co.

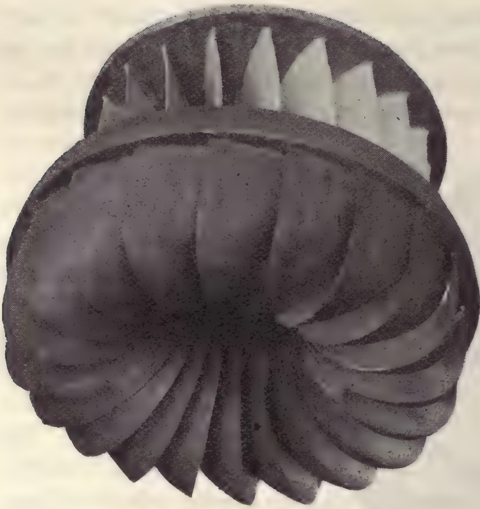


Fig. 138.—Victor or "Type A" Runner of The Platt Iron Works Co.

Fig. 139.—High Head or "Type B" Runner of The Platt Iron Works Co.

The Smith-McCormick runner is manufactured by The S. Morgan Smith Company. This company has also recently brought out a new wheel called the "Smith Turbine," of greater power and higher speed, the runner of which is illustrated by Fig. 137. Fig. 138 represents the Victor runner or "type A" runner of The Platt Iron Works Company, designed for low heads.

Figure 139 is the "type B" runner, of the same company, designed for medium heads. This runner again illustrates the tendency to return to the earlier forms of runner for medium head wheels. This latter type has also been adopted by other manufacturers of turbines, as may be seen by reference to Fig. 140 which shows a 1,250 H. P. single turbine runner and the twin runners of a 2,400 H. P. double turbine manufactured by the Allis-Chalmers Company. Fig. 141 shows a bronze runner for a horizontal double quarter-turn discharge setting, to develop 8,500 H. P. at 300 R. P. M. under 450 feet head built for the Grace, Idaho, plant of the Telluride Power Company by the Allis-Chalmers Company. Fig. 142 shows one of the 10,000 H. P. runners for the Keokuk plant of the Mississippi

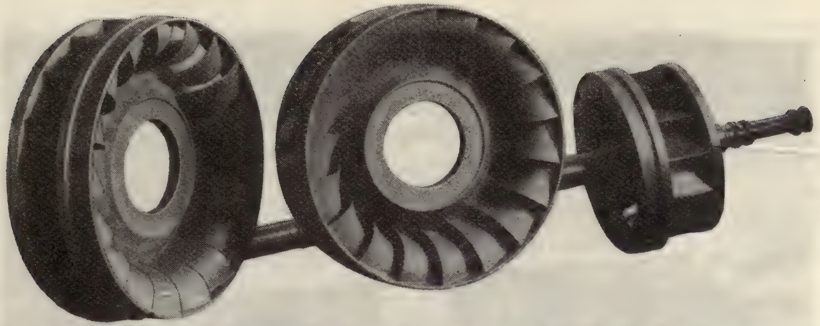


Fig. 140.—1250 H. P. Single Turbine Runner and 2400 H. P. Double Runner.

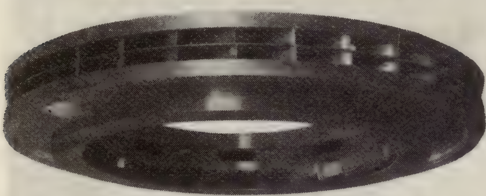


Fig. 141.—Bronze Runner for 8500 H. P. Horizontal Double Quarter-turn-discharge Reaction Turbine for Grace, Idaho, Plant of Telluride Power Co.

River Power Company built by the Wellman, Seaver, Morgan Company. This runner is 138 inches measured on the medial diameter of the vanes and 144 inches in diameter at the band.

Figure 143 is from a shop photograph of the Shawinigan Falls turbine manufactured by the I. P. Morris Company. This is one of the largest turbines ever constructed and develops 10,500 H. P. under a head of 140 feet. It is a double mixed inflow type with spiral casing and a double draft tube through which the water discharges outward from the center. The diameter of the casing at the intake is ten and one-half feet and the sectional area gradually diminishes around the wheel in proportion to the amount of water flowing at each point. The wheel complete is thirty

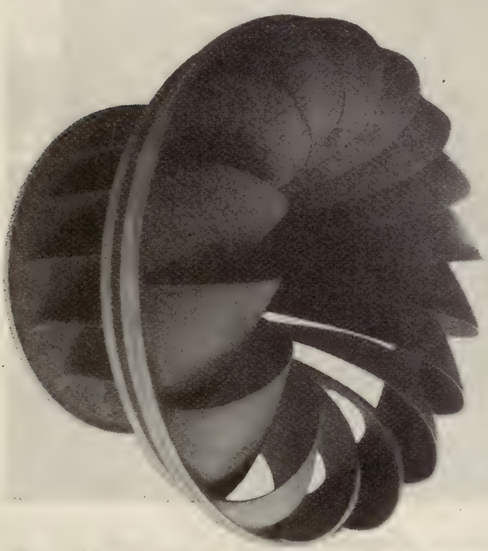


Fig. 142.—10,000 H. P. Runner for Keokuk Plant.

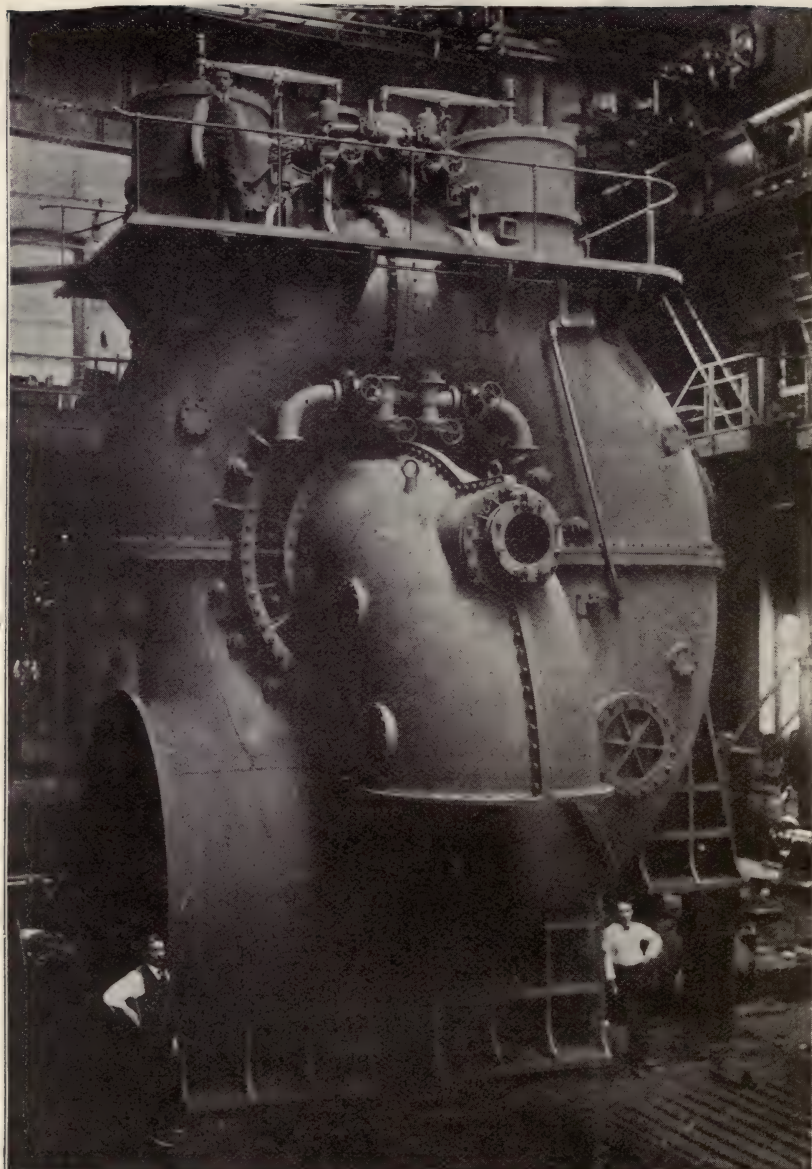


Fig. 143.—Shawinigan Falls Turbine, Manufactured by I. P. Morris Co. (see page 231).

feet in height and weighs 182 tons. The runner, which is of bronze, is shown in Fig. 144.

Figures 145, page 234, and 146, page 235, show two sections of a single turbine of the Francis inflow type built for the Snoqualmie Falls plant of The Seattle & Tacoma Power Company by The Platt Iron Works Company. The turbine has a capacity of about 9,000 H. P. under 270-foot head at 300 R. P. M. The runner is sixty-six inches in diameter and has a width of nine and one-half inches measured radially through the buckets.*

For further details see Figs. 168, 174 and 175, pages 259, 266, 267.

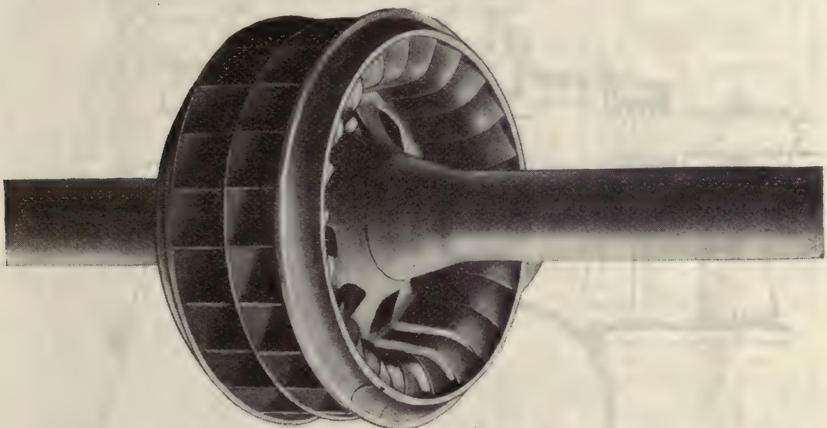


Fig. 144.—Shawinigan Falls Turbine Runner.

117. **American Impulse Wheels.**—Some of the many forms of American buckets used are shown in Fig. 16, page 17, with the approximate date of their invention or design.

The general arrangement of a double 2,000 H. P. unit, running at 200 R. P. M. under 500 foot head is shown in Fig. 147, page 236. This is one of three units installed by The Pelton Water Wheel Company for The Telluride Transmission Plant of Colorado.

The wheels are of cast steel fitted with steel buckets, held in position by turned steel bolts. They are connected by a flexible coupling to a 1,200 H. P. generator.

Figure 148, page 237, shows the runner of an impulse wheel made by the same company. This is nine feet ten inches in diameter, and is designed to develop 5,000 H. P. at 225 R. P. M. under an effective head of 865 feet.

* See "Engineering News" March 29, 1906.

Figure 150, page 239, shows the runner of an impulse wheel manufactured by the Abner Doble Company. This runner was from the Doble Water Wheel Exhibit at the St. Louis Fair and developed 170 H. P.

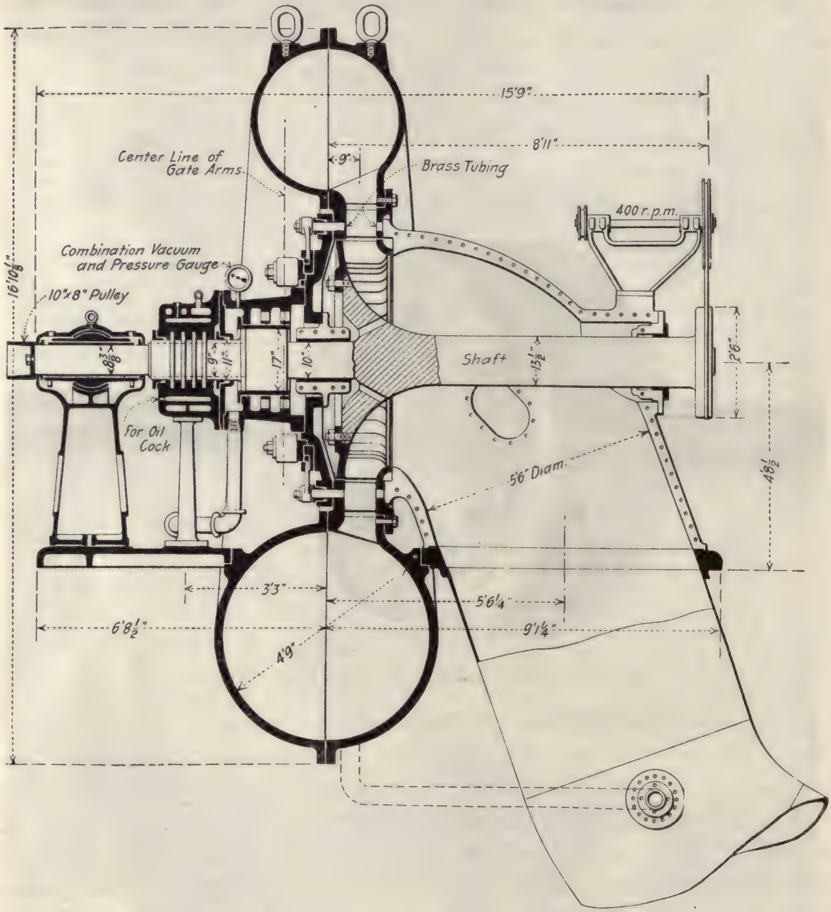


Fig. 145.—Section Snoqualmie Falls Reaction Turbine. The Platt Iron Works Company (see page 233).

at 170 R. P. M. under a head of 700 feet and generated direct current for use on the intramural railway.

Figure 149, page 238, shows the runner of an impulse wheel of 5,500 H. P. under 865 feet fall at 250 R. P. M. built for the Edison Electric Company's Kern River Plant No. 1 by the Allis-Chalmers Company.

In addition to the tangential wheels already described, a few manufacturers have developed wheels of the Girard type. One such wheel, designed and built by The Platt Iron Works Company, is illustrated in Figs. 151 to 154, pages 240 to 242 inclusive. Fig. 151 is a section-elevation showing the arrangement and design of the guides and buck-

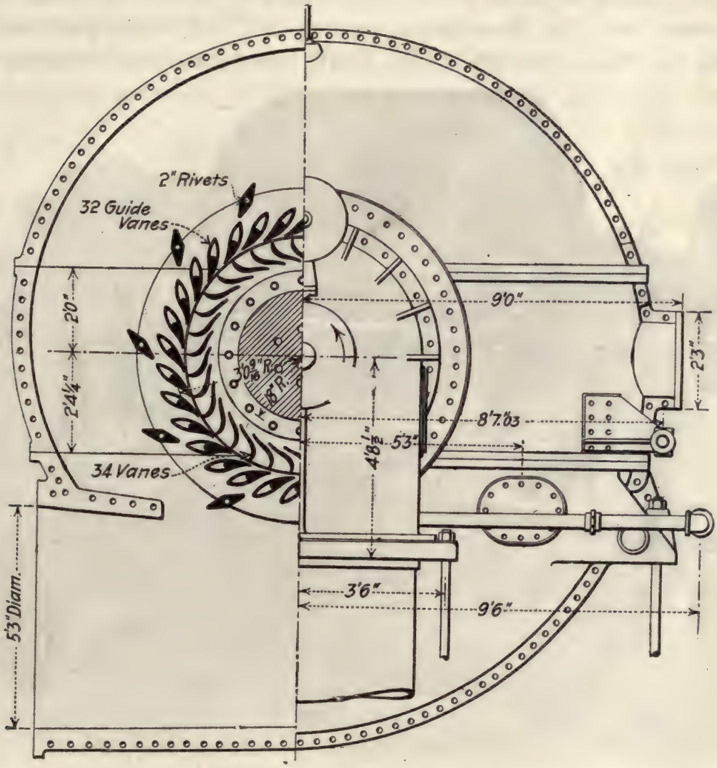


Fig. 146.—Section-Elevation Snoqualmie Falls Reaction Turbine. The Platt Iron Works Company (see page 233).

ets of the wheel. Fig. 152, page 241, shows a section through the wheel and on the line of the shaft. In these figures *W* represents the runner; *B B* the buckets; *g*, the inlet guides, and *G*, the gate by which all or a portion of the guide passages may be closed and the power of the wheels reduced. The gate *G*, is connected by the gearings *Gr*, with the rod *r*, which is connected through the rocker arm with the governor mechanism. The wheel or runner of this turbine is shown by

Fig. 153, page 242, and a general view of the wheel is shown by Fig. 154, page 242.

118. Modern Turbine Development.—Modern European turbine practice has been the development of the last thirty years. European manufacturers approached the subject more on the basis of theoretical analysis than was at first done in America. The conditions of development have also been largely special and not under such uniform conditions as in America. The result has been the development of

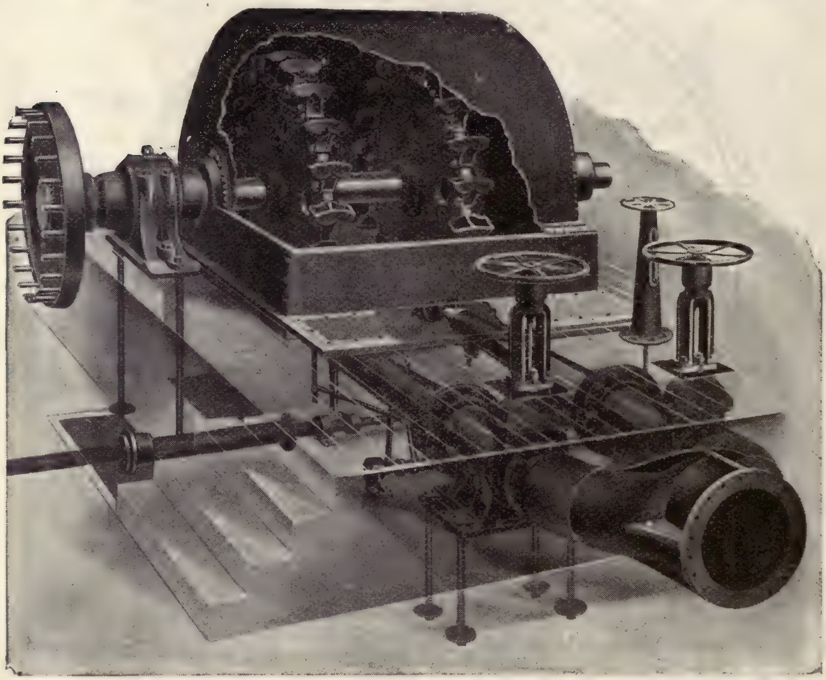


Fig. 147.—Telluride Double Tangential Wheels. 2000 H. P. 500 Foot Head. Pelton Water Wheel Company (see page 233).

special designs for special locations and the rapid accumulation of a considerable experience under a wide range of conditions. While the radial flow turbines were the earlier type developed, European practice has been largely centered on the axial flow wheels of the Jonval type for complete turbines, and axial flow and radial flow wheels of the Girard type for partial turbines under high heads.

The axial flow turbine while simple in construction and low in cost is difficult to regulate and hence the demands of electrical development for close regulation has given rise to a variety of modern European de-

signs which were summarized by Mr. J. W. Thruso (in 1905) essentially as follows: *

First: For low heads to twenty feet. Radial inward flow, reaction turbines with vertical shafts and draft tubes.

Second: For medium heads, 20 to 300 feet. Radial inward flow reaction turbines with horizontal shafts and concentric or spiral cases and draft tubes.

Third: For high heads over 300 feet. Radial outward flow, full or partial action turbines (of the Girard type) with horizontal shafts, often with draft tubes; also, modified impulse wheels of a tangential type.

Fig. 148.—Pelton Tangential Water Wheel Runner. Designed for 5000 H. P. at 865 foot head and 225 R. P. M. (Pelton Water Wheel Co.).

The types of turbines for low and moderate heads are modifications of the Francis inward flow turbine.

In the United States an enlightened demand for true economy in water power installation has materially improved the results that can now be secured from hydraulic turbines.

The "Standard" wheel in its original form finds little market, except in the small mill powers where economy is unrecognized or unnecessary, and in many cases the manufacturers of such wheels have discontinued business. Many former manufacturers of standard wheels have improved their turbines, and by more or less special designs have adapted them to the special conditions to which their characteristics apply. They have added other types in order to extend the range of application, and while still adhering to a system of construction which admits of economy in the cost of manufacture, adapt their design to the special condition of each installation. Other manufacturers give

* See "Modern Turbine Practice" page 3, by J. W. Thurso. D. Van Nostrand Co., New York, 1905.

attention only to special designs for the larger installations where the importance of the results admit of special treatment and the necessary higher first cost. There has been an exceedingly rapid improvement in the character of turbine installations, in the direction of higher efficiency, greater strength and durability and application to a wider range of conditions. While guarantees above eighty per cent. were rare prior to 1905, guarantees of eighty-six per cent. to eighty-eight per cent. are now readily secured for high grade installation, and the importance of first class results are being recognized by often including

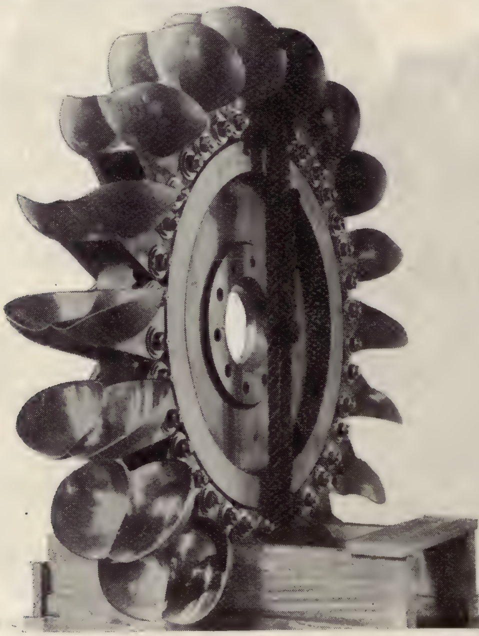


Fig. 149.—Impulse Wheel, 5500 H. P. 865-Foot Head, 250 R. P. M., Built by the Allis-Chalmers Co. (see page 234).

a bonus and penalty clause for the payment of a stipulated sum per unit for each per cent. above or below the contract guarantee.

The increase in the specific power of wheels and the ability of manufacturers to construct wheels of larger diameter, has materially altered the best practice for low heads by the adoption of single direct connected units which, while greater in expense on account of lower speeds, have many mechanical advantages over the higher speed of the

installation having two or more runners. Taylor summarizes the advantage of the single runner unit as follows: *

1. "Only one gate mechanism is required, and this is located above the head cover of the turbine and is accessible at all times for inspection while the unit is in operation. The only parts of the turbine that

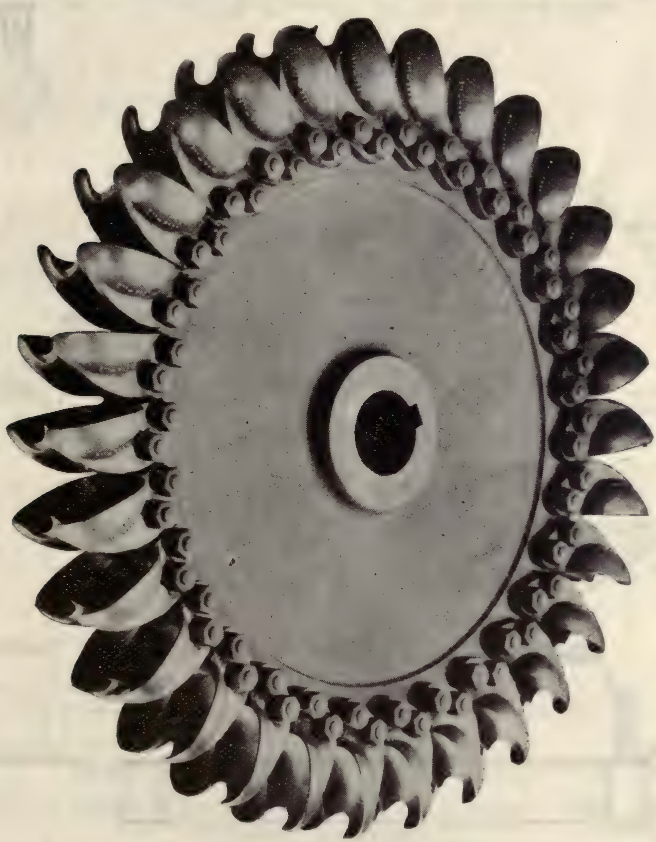


Fig. 150.—Doble Runner. Pelton Water Wheel Company (see page 234).

are submerged are the runner and the guide vanes. Repairs can be made to the gate operating mechanism without dismantling the turbine.

2. "Owing to the fact that only one gate operating mechanism is used, involving a small number of parts, the chance for breakage is

* From extracts from a paper before the Canadian Society of Civil Engineers, Jan. 15, 1914. See Engineering News, February 5, 1914.

reduced to a minimum, and lost motion and deflection in the governor engine connections are avoided.

3. "It is possible to secure an ideal draft tube of long flaring section, without an obstruction or sudden turn. Therefore it is possible to use runners of the very highest specific speed, as the draft tube can

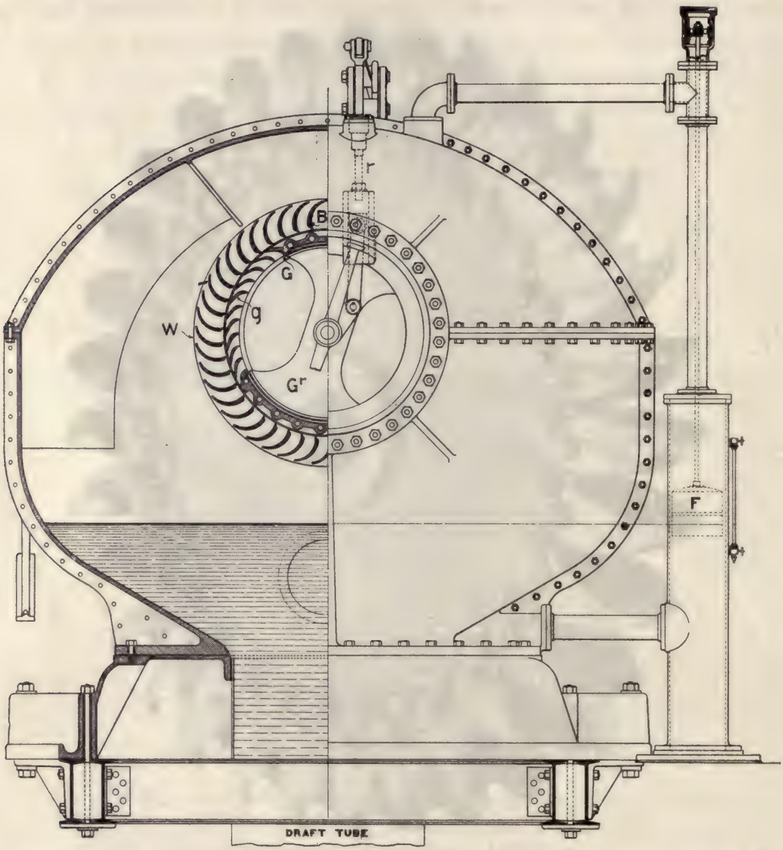


Fig. 151.—End Section and Elevation Girard Impulse Turbine With Draft Tube. Platt Iron Works Company (see page 235).

be designed to convert the velocity at the discharge from the runner buckets into effective head with small degree of loss.

4. "It is possible to mold in the concrete a spiral turbine casing similar in design to the cast-iron spiral casings used in connection with high-head turbines. It would be impracticable to prepare spiral cas-

ings for vertical or horizontal turbines having two or more runners, for obvious reasons.

5. "It is very often possible to locate the runner and gate mechanism above high tailwater level, so that after closing down the head gates and draining the wheelpit, an attendant may examine all parts of the turbine without first having to pump out the wheel-chamber."

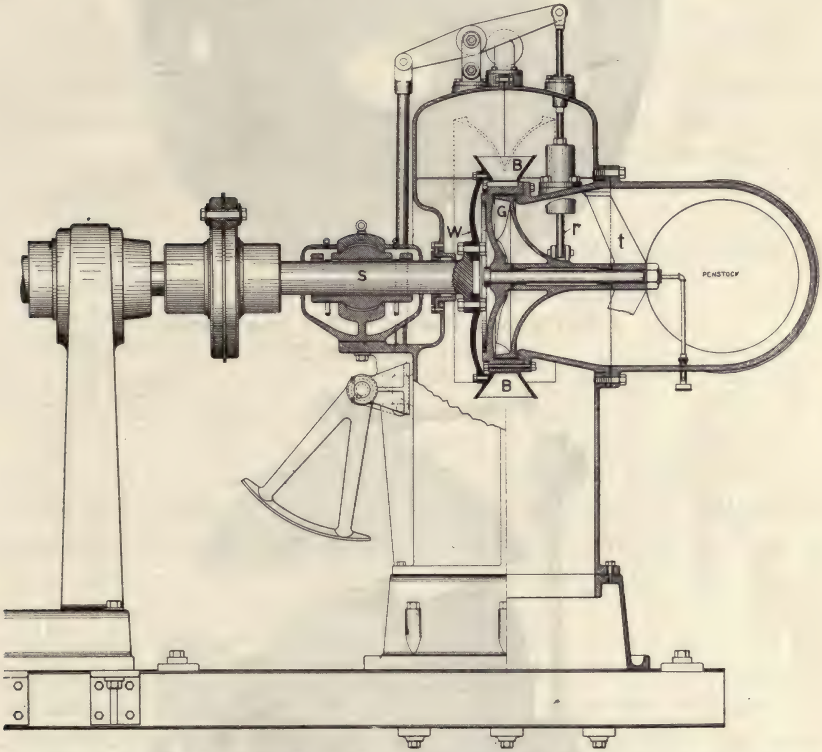


Fig. 152.—Longitudinal Section Girard Impulse Turbine. Platt Iron Works Company (see page 235).

The improvement in the design of reaction runners has greatly reduced or eliminated the corrosion formerly due to imperfect hydraulic conditions, and has extended their use to heads as high as 670 feet. The types of wheels now generally installed in the United States are confined to reaction turbines for heads as high as 700 feet and tangential wheels for still higher heads.

Fig. 153.—Runner of Girard Turbine. Type C, High-Pressure Runner. Platt Iron Works Company (see page 235).

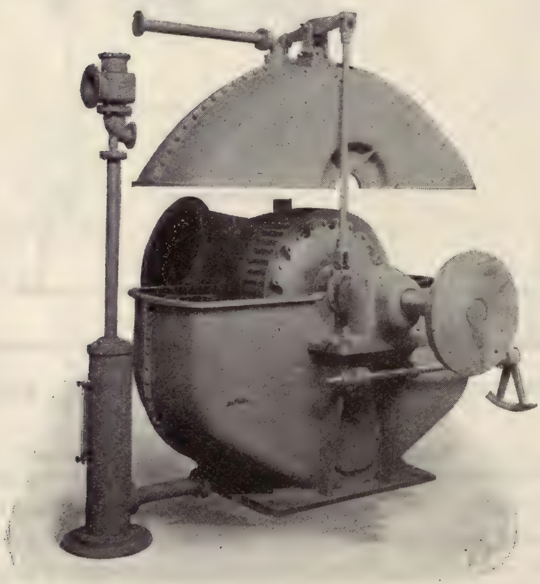


Fig. 154.—General View of Girard Turbine with Cover Raised. Platt Iron Works Company (see page 235).

CHAPTER X

TURBINE DETAILS AND APPURTENANCES

119. The Runner—Its Material and Construction.—The runners of most reaction turbines (see Figs. 122, 124, 132, 136 to 140) consist of hubs, crowns and rings, to which the buckets are attached. The wheels are sometimes cast solid, and sometimes built up. In built-up wheels the buckets are first cast, or otherwise formed, after which they are placed in form or mould and the crowns, hubs and rings are cast to them.

There are four materials in general use in the construction of water wheel runners, namely:

Cast Iron
Cast Bronze
Cast Steel
Plate Steel Vanes, cast in hubs of cast iron.

The best material to be used depends upon the local conditions of head, speed and quality of the water. The use of bronze is necessitated by requirements for resistance to corrosive chemical action. In general, manufacturers prefer the cast bronze runner because of the greater strength of casting and greater ease in finishing, although the cost is greater. Cast iron runners have been generally used under relatively low head conditions, but their use in this regard is extending. Their principal advantage lies in the fact that the use of cast iron results in more homogeneous castings and greater stiffness. Cast steel is tougher and stronger than cast iron, but castings are liable to show blow holes and cavities and a more uneven surface. An advantage in the use of cast steel is the less liability to cracks from shrinkage stresses and from handling.

Probably the majority of cast wheels manufactured at the present time are made in one solid casting of buckets, rings, hubs, and crowns. The buckets are formed by carefully prepared cores and in such manner as to leave them uniform in spacing and thickness, and smoothly finished so as to admit of the passage of water through or between them without excessive friction. With wheels so cast, no material finishing or smoothing of the surfaces of the bucket is

practicable, and the casting must come from the sand with a satisfactory surface. In wheels cast solid, great care is necessary in order to prevent serious shrinkage strains. This is practically overcome by the use of soft iron, which results, however, in increased wear of runners when subjected to the action of sand-bearing waters.

Turbine water wheels for low heads, are usually made of cast iron or cast iron with steel buckets. Wheels for high heads are made of cast iron, cast bronze or of cast steel (see Figs. 141 and 144, pages 231 and 233). A higher surface finish of the bucket is possible with buckets cast separately, but when separate buckets of cast iron are made and afterwards united, the runner must be strongly banded in order to give it the necessary strength, since there is a tendency for the buckets to work loose, due to the imperfect bond between the cast bucket and the hub casting. Their use has not proved entirely satisfactory either in Europe or America, except under low head conditions, and where the vibration is not great. Steel plate buckets cast welded into cast iron hubs and discharge rings are much more satisfactory within certain limits of size. When properly manufactured there is little tendency for the buckets to work loose. The union of the buckets and hub of a built-up turbine runner of this kind manufactured by the Allis-Chalmers Company is shown in Fig. 155, where under test the bond obtained between the steel plate buckets and the cast iron hub was so great as to cause the plate to tear away without pulling loose from the casting. For high head conditions, and for runners above six feet in diameter, the built-up runner must usually be replaced by solid cast runners on account of strength and stiffness.

Fig. 155.—Result of Strength Test of Allis-Chalmers Built-up Runner.

The abrasion of turbine runners is due to the mechanical erosive action of sand or other material carried in the water. Runners of

bronze will not resist this action as well as runners of cast iron or cast steel. With thin plate steel buckets an equal depth of abrasion is a larger proportion of the total thickness than is the case in cast buckets which usually have greater thickness. Cast steel runners are of great strength and will in general stand more severe usage than runners of other materials. Fig. 156 shows a cast iron runner which was set in a vertical position and which, while in operation, dropped from the shaft and revolved on the bridge tree, breaking the lower ends of the buckets as shown. Fig. 157 shows a runner with plate steel buckets, which was injured as shown by a steel bar working loose from the casing and falling against the revolving runner. It is not feasible to design runners which will be proof against such extraordinary accidents, but the design of the casings, fastenings, and supports should be such as to prevent the occurrence of such accidents so far as practicable.

Fig. 156.—Vertical Cast Iron Runner Injured by Falling From Shaft.

Fig. 157.—Steel Plate Runner Injured by a Steel Bar Which had Become Loose From the Casing.

Up to about 1905, the reaction turbine was not considered as suitable for heads above 300 feet. This was due to the fact that, especially under high heads and consequent high velocities, reaction runners were rapidly destroyed by corrosion or pitting, which usually occurred on the back of the bucket. It is now well established that corrosion is generally due to improper hydraulic design, and to obviate this action the turbine must be so designed as to avoid sharp curves, improper contractions or depressions in the bucket surface, and so that the bucket will be completely

filled under all conditions of gate. If the design is such that under some or all conditions of gate the water leaves the back surface of the bucket, a vacuum is created and oxygen is released from the water, causing rapid oxidation. The oxide so formed may be removed by the current of water where the bucket is filled under other gate conditions, so that the action may be quickly repeated and the corrosion may proceed with great rapidity. This action takes place with heads as low as thirty feet with runners of improper design, and has been prevented under heads as high as 600 feet or more by proper design.

Corrosion may take place with wheels of high efficiency as the efficiency is less affected than the conditions giving rise to corrosion by slight errors in design; besides, a runner may be well designed for certain gate openings and still allow the occurrence of conditions which will bring about corrosion under other conditions of gate. It is therefore especially desirable to operate high head plants as near the point of maximum efficiency as possible in order to minimize this effort. No material is free from this action, but when corrosion is due to acids in the water, bronze runners will usually give better results than runners of iron or steel.

The runners of Girard impulse wheels (see Fig. 154, page 242) are made in the same manner as reaction runners.

The runners of tangential wheels are usually made with separate buckets and body (see Figs. 149 and 150, pages 238 and 239). The bodies are made, according to the severity of the service, of cast iron, semi-steel, forged steel, etc. The buckets, dependent on the conditions of service, may be of cast iron, cast steel, gun metal, bronze, etc. The buckets, in the best wheels, are cast, shaped and polished and carefully fitted to the wheel body. The bolt holes are then carefully drilled and reamed and the buckets are bolted in position by carefully turned and fitted bolts.

120. Diameter of the Runner.—The diameters of reaction runners are measured at the inlet, and, when the buckets at the inlet are parallel and of one size, the determination of the turbine diameter is a simple matter (see Fig. 158, page 247, diagram A). In order to give the runner greater speed and capacity, the buckets are cut back from the crown to a point opposite the bottom of the gate opening (see diagram B). In such cases the edges of the buckets are sometimes made parallel with the shaft but are usually inclined. In the latter case, the diameter of the wheel at its top may be considerably reduced over its diameter at the offset. In such cases the cutting

back of the runner may be one or more inches at the bottom line of the gate with an inch or more inclination to the top of the buckets, and the diameter of the wheel at D and D''', diagram B, may differ from two to six inches or even more.

With wheels so constructed, there is considerable difference in the practice of different manufacturers in measuring and listing the diameter of the wheels made by them. In some cases, the inside diameter, from ring to ring D, diagram B, of the runner, is taken as the nominal diameter. In other cases, the diameter is taken at the inner angle of the offset as D'. In a number of cases the diameter is measured on the pitch diameter at about the center of the gateway D'', and in other cases, the diameter is measured at the upper and smaller diameter of the runner D'''. This variable practice leads to a considerable difference in the nominal diameter of the

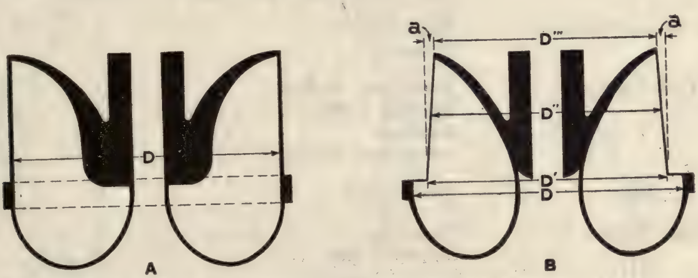


Fig. 158.—Positions at Which Nominal Diameters are Measured (see page 246).

various turbines as listed in the catalogues or specified in proposals and frequently a runner of a certain specified diameter of one manufacturer may be two to six inches larger than the runner of another manufacturer which is specified as of the same diameter. This discrepancy in the method of measuring the diameter of turbine runners may account for the apparent greater capacity, higher speed or greater power of the wheels of one manufacturer over those of another.

The practice of some of the American manufacturers of turbines, in measuring and specifying the diameters of their wheels, is shown in Table 25. In this table, all runners which are not cut back and with edges parallel to the shaft, are classified as Style A, even where they differ widely from the form shown in diagram A, Fig. 158.

All runners with buckets cut back are classified as Style B, even where the bucket edges are parallel with the shaft.

The diameters of tangential runners are usually measured between the centers of buckets or on the diameter of the circle on which the center of the jet impinges on the buckets (see Fig. 159, page 249).

TABLE 25.

Practice of Various American Manufacturers in Measuring and Cataloging the Diameter of Turbine Water Wheels.

Manufacturer.	Name of Runner.	Style.	Point of Measurement.
Allis-Chalmers Co.	....	B	D″
Dayton Globe Iron Works	American	A	D
	New American	A	D
	Special New American[1]	B	D′
	Improved New American[2]	B	D′
Rodney Hunt Machine Co.	McCormick[3]	B	D
	Hunt	A	D
James Leffel & Co.	Standard Leffel	A	D
	Special Leffel	A	D
	Samson	B	D
	Improved Samson	B	D
J. and W. Jolly	McCormick	B	D″
Platt Iron Works Co.	Type A	B	D″
	Types B and C	A	D
S. Morgan Smith Co.	McCormick[4]	B	D′
	Smith	B	D″
The Trump Manufacturing Co.	Standard Trump[5]	B	D″′
	High Speed Trump[1]	B	D
Wellman, Seaver, Morgan Co.	....	B	D″
I. P. Morris Co.	....	B	D″
Hydraulic Turbine Corp. (Camden Water Wheel Co.)	....	B	D″

[1] Bucket of high speed runner has parallel edges but is cut back as shown in B.

[2] Fillet at angle. Diameter measured just above.

[3] Diameter of Hunt-McCormick runner is measured at the crown which projects beyond the tips of the buckets and is essentially the same in diameter as at D′.

[4] Diameter of the Smith-McCormick runners is measured at the crown which projects beyond the tips of the buckets and is essentially the same in diameter as at D′.

[5] Diameter at D is 20% greater than at D″′.

121. The Details of the Runner.—The reaction runner will vary in design with the conditions under which it is to operate and the experience and ideas of its designer. In American practice the manufacturer of standard wheels usually constructs a series of runners of similar homologous design; that is to say, each wheel of the series has all of its dimensions proportional to that of every other wheel of the series, and is of similar design in all of its parts.

On account of demands for considerable variations in speed or power, or on account of improvements which have been found de-

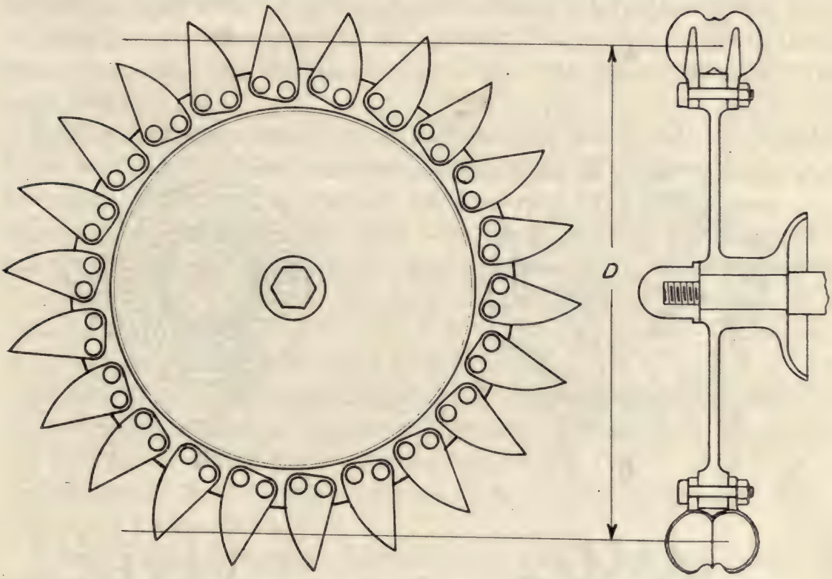


Fig. 159.—Measurement of the Diameter of the Tangential Wheel.

sirable by reason of the demands of his trade, the manufacturer often designs and constructs several series of wheels, each of which is particularly adaptable to certain conditions which he has had to meet (see Tables 22 and 23, page 222). In such cases each series is best suited for the particular condition for which it was designed and is not necessarily obsolete or superseded by the later series.

The curves of the runner buckets (see Figs. 13, page 11, 14, page 12, 119, page 217, 120, page 219, 122, page 221, 128–130, pages 224 and 225 and 160, page 250) must be such as to receive the jet of water from the nozzle or guides without shock, permit it to pass

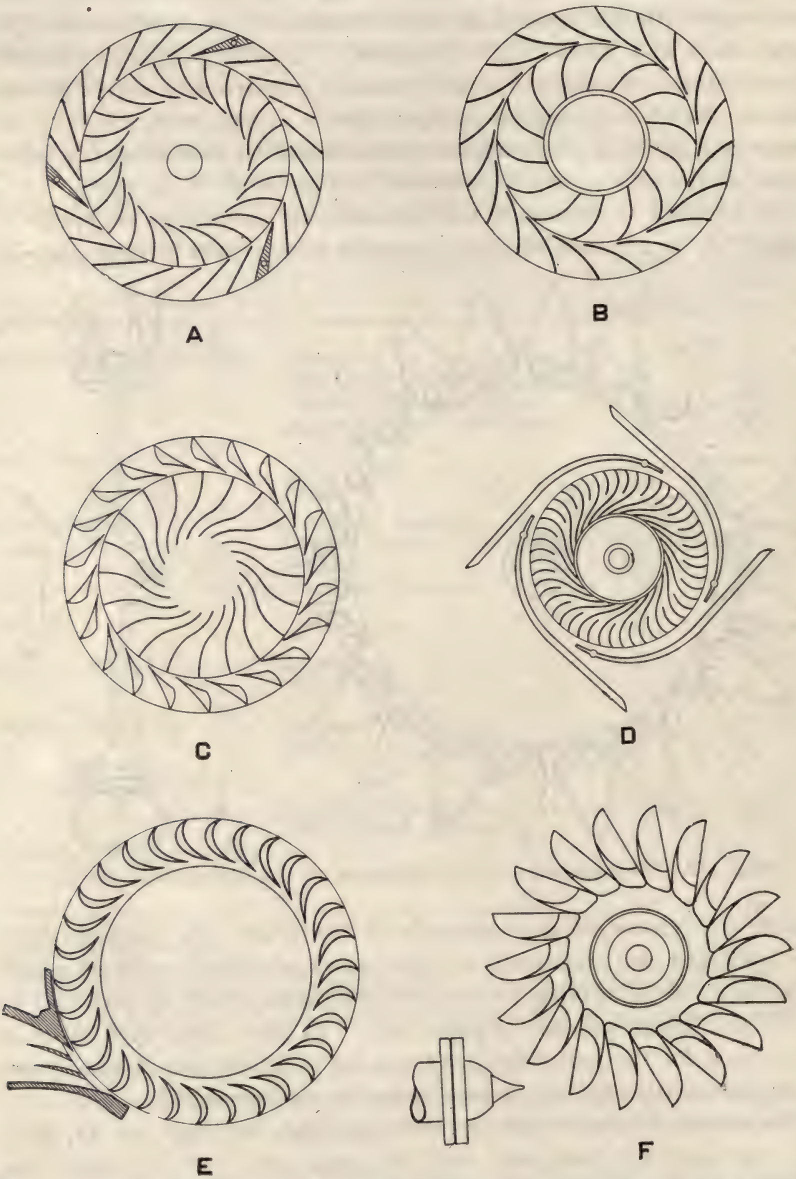


Fig. 160.—Curves of Buckets and Guides in Turbine Wheels (see page 249).

along the surface of the buckets or through the passages in the runner with minimum friction, and discharge it as nearly devoid of velocity as practicable.

To accomplish this, the relative position and relation of the curves of guides and buckets must be carefully designed. As the jet of water is always directed forward in the direction of the revolution of the wheel, the jet has an original velocity in that direction, and, since the bucket must be so shaped as to give a continued contact, as the jet progresses and the wheel revolves, the portion of the bucket farthest away from the guides must be curved backward, and terminate at such an angle as will permit the jet to pass away from the wheel with free discharge (see Figs. 160, page 250 and 114, page 206).

Reaction runners are made either right or left handed as conditions demand. Where turbines are installed in pairs on the same horizontal shaft, a right and left hand wheel is commonly used discharging into the same draft tube (see Figs. 330 and 334, pages 521 and 524). When looking at the top of the runner, if the wheel is designed to move in the direction of the hands of a watch, it is called a right handed wheel, and if it moves in the other direction, it is called a left handed wheel (see Fig. 161).

The buckets, hub, crown, and ring of the reaction runner must be of sufficient strength to receive the impact or pressure of the moving column of water under the working head, and to transmit the energy to the shaft through which it is to be transmitted to the machinery to be operated.

A heavy ring is usually desirable, both to give strength and support to the outer edge of the buckets and also, under some circumstances, to give the effect of a flywheel in order to materially assist in maintaining uniform speed. Floating blocks or other material, in spite of the use of trash racks, sometimes reach the turbine, and when caught between the buckets and the case are apt to cause serious injury to the buckets.

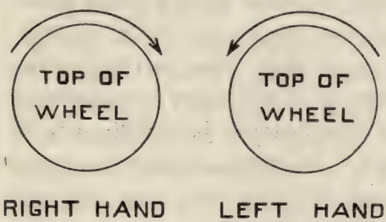


Fig. 161.—Direction of Rotation of "Hand" of Water Wheels.

The runner is attached to a shaft passing through the hub, to which it should be closely fitted and strongly keyed to prevent its becoming loosened by vibration and

the strain of operation. This is especially necessary in vertical wheels, for if, under these conditions, the wheel becomes loosened and drops from the shaft, it is apt to be practically destroyed (see Fig. 156, page 245). Impulse runners acting under high heads are subject to heavy shocks and must be especially substantial.

122. Vertical Turbine Bearings.—In all turbines where the discharge is axial and only in one direction, there is a reaction in the other direction that tends to unbalance the wheel and to cause a thrust in the direction opposite to the discharge. The leakage into the space back of the runner frequently produces a thrust in the opposite direction which may be wholly or partially relieved by openings left in the runner, usually close to the axis. In large units an attempt is made to balance these various pressures with some form of thrust bearing to sustain the difference in pressure which will occur under different conditions of operation.

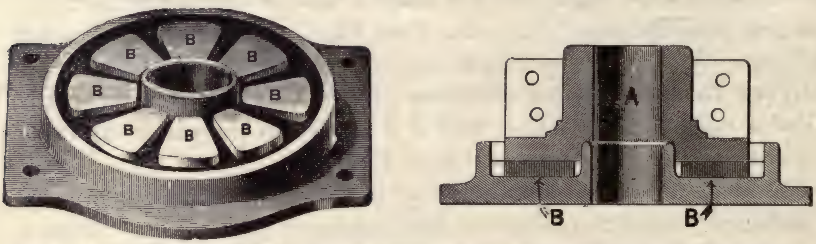


Fig. 162.—Geylin (Patent) Glass Suspension Bearing (R. D. Wood & Co.).

In many small single vertical turbines a simple step bearing is used. The bearing itself in American turbines usually consists of a lignum vitae block, turned to shape, and centered in a bearing block which is held firmly and centrally in place by the cross trees. The bearing block is shown by *T*, and the cross trees by *t*, in Figs. 128, page 224, 129, page 225 and 172, page 264. The bearing on the shaft itself is usually a spherical sector, or some other symmetrical curve of similar form. In some cases this bearing is cut directly in the shaft itself (see Fig. 129, page 225). In others, a cast iron shoe is provided and attached to the shaft (see *M*, Figs. 127, page 224 and 171, page 263). Above the turbine a second bearing is also provided (see *T′*, Figs. 127 and 129) to keep the shaft in vertical alignment. This bearing in American wheels is usually of the type shown in Fig. 169, page 260, except that it is adapted to its vertical position.

In the Geylin-Jonval turbine, manufactured by R. D. Wood Com-

pany, a patent glass suspension bearing is used (Fig. 162, page 252). This bearing is attached above the wheel (see *T*, Fig. 121, page 220) and has the advantage of being readily accessible. The turbine is here suspended on a circular disc composed of segments of glass *B B* (Fig. 162), arranged with depressed divisions which form a continuous space around each segment of which the disc is composed, allowing, while the turbine is in motion, a perfect, free circulation of the lubricating matter with which the space is filled.* The bearing is a true metallic ring *A*, firmly secured to the turbine shaft which revolves on these stationary glass segments.

The above bearings are applicable only to small vertical units. In modern vertical installations of large capacity, much more elaborate and complicated bearings are necessary. These large bearings have given much trouble in the past, but the experience of the last twelve years (1903–1915) has resulted in the development of some very satisfactory designs.

At the power plant of The Niagara Falls Power Company a thrust or hanging bearing of the disc type was installed about 1902 (see Fig. 163, page 254). In this bearing the shaft is suspended to a revolving disc carried on a stationary disc. The discs are of close-grained charcoal iron of 25,000 pounds tensile strength and of fourteen inches inside, thirty-four inches outside diameter. The lower or fixed disc is dowelled to a third disc with a spherical (three feet four inches radius) seat. This is to provide for an automatic adjustment for slight deviations from the vertical due to uneven wear of the discs and other causes.

The bearing surfaces between the discs are grooved to allow a circulation and distribution of the oil over the surface.

Three methods of lubrication,—forced, self, and a combination system, were experimented with and the combination system finally adopted. In the system of forced lubrication, the oil enters the fixed disc at two diametrically opposite points and is forced between the discs under 400 pounds pressure. Self-lubrication is accomplished by oil supplied at the inner circumference of the disc and thrown outward by centrifugal force.

The disc bearings are enclosed in a case provided with sight holes through which the condition of the bearing as well as the temperature of the oil can be observed. A thermometer and an incan-

* Catalogue of R. D. Wood & Co., 1901, page 107.

descent light are suspended in the casing for this purpose. The oil is cooled by water circulating pipes inside the casing.

The shaft is provided with a balancing piston (see Fig. 164, page 255) supplied with water from a pipe entirely independent of the penstock and under a head of 136 feet. This piston carries the greater

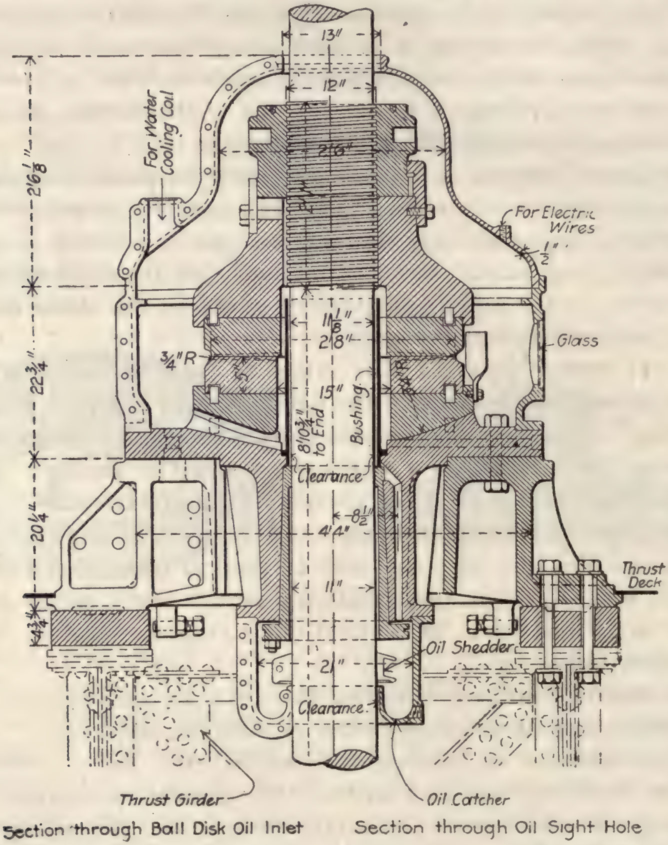


Fig. 163.—Vertical Thrust or Hanging Bearing of the Niagara Falls Power Co. Eng. Record, Nov. 28, 1903 (see page 253).

part of the load, less than two per cent. of the load being left to be carried by the oil-lubricated disc bearing described above.

Figure 165 shows a section of the combined roller and pressure disc bearing used on the 10,000 H. P. wheels of the Mississippi River Power Company. It is placed just above the turbine and below the direct-connected generator (see page 256).

A type of thrust bearing has recently been developed by Mr. Albert Kingsbury which is highly efficient, showing a coefficient of friction less than .001 as an average. This bearing is an oil thrust type in which the adhesion of the oil to the moving surface is utilized to lubricate the bearing. The moving or rotating surface of the bearing consists of a smooth runner ring or washer; the stationary part, over which the runner moves, consists of babbitted shoes, each shaped like the sector of a ring. The construction is shown in Fig. 166, page 257. Each shoe is supported by a hardened steel pivot pin, placed slightly off center in the direction of rotation and free to move slightly, allowing the film of lubricating oil between the shoe and the runner to take the form of a thin wedge. The bearing is designed to run submerged in oil at all times.

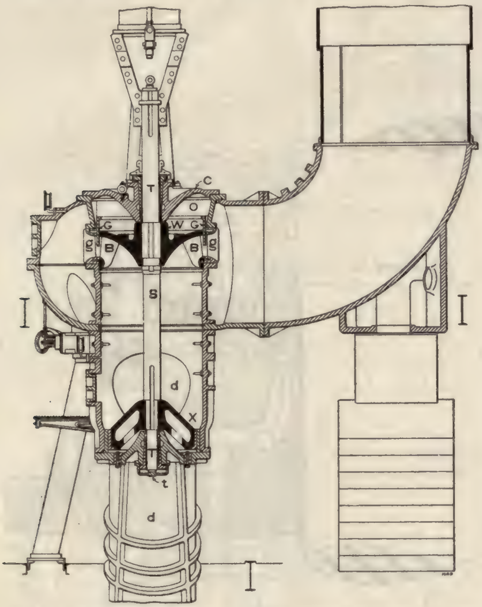


Fig. 164.—Section of Turbine Used in New Power House of The Niagara Falls Power Company, Showing Balancing Hydraulic Piston Used to Sustain Turbine and Shaft. Eng. Record, Nov. 28, 1903 (see page 254).

The runner N is attached to the shaft, rotates with it and slides upon the stationary shoe B. These shoes are sectors of a ring and each sector is supported against the thrust by the pieces D and E. E is crowning on the face to permit the shoe to adjust itself to the best bearing condition. An adjusting wedge permits the shoes to be adjusted for an equal division of the load. The base ring C is supported on a spherical leveling washer or on a flat base. The bearing is contained in a housing which is kept full of oil to a level above the sliding surface.

Tests of this device under very severe conditions have been made, and the results are said to have been very satisfactory. The safe bearing load on this type is apparently considerably greater than has been considered safe heretofore. A thrust bearing for a vertical

hydro-electric generating unit containing six shoes of forty-eight inches outside diameter, with unit pressure of 350 pounds per square inch under test at a mean surface speed of 900 feet per minute, showed a coefficient of friction of .008 with temperature rise of three degrees Centigrade. This bearing had operated about a year without requiring attention.

In most European vertical turbines, even of small sizes, the step bearing is simply a guide, the main bearing being above the turbine and readily accessible.

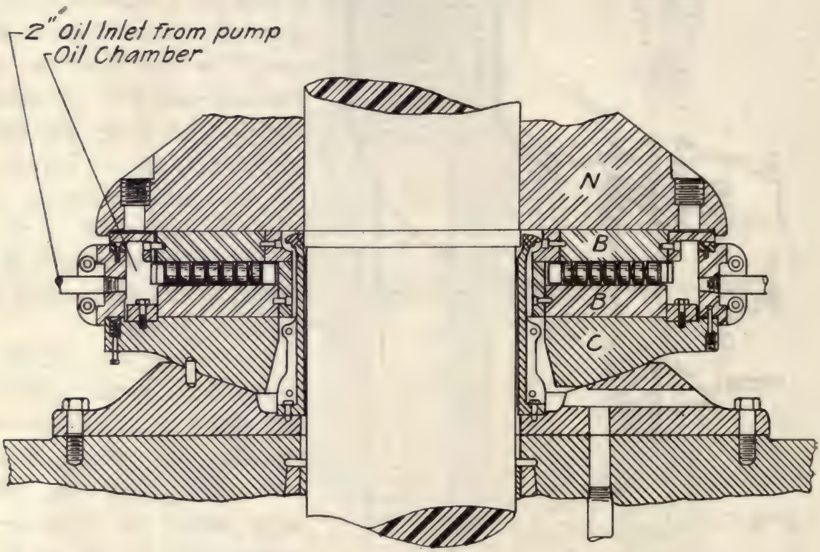


Fig. 165.—Combined Roller and Disc Bearing of the Mississippi River Power Co. (see page 254).

Figures 167 and 168, pages 258 and 259, represent vertical bearings of this kind. In these bearings *C* is a spherical sector so arranged as to take up any slight error in the vertical alignment of the shaft. Fig. 167 is a ball bearing; the hardened steel balls *AA*, revolve between the special bearing plate *B* and B_1.

In Fig. 168 oil is pumped under pressure through the inlet pipe, *OE*, into the space *A*. By its pressure the bearing plate *B*, is raised from its companion plate *B*, and the oil escaping between the plates lubricates them and overflows through the overflow pipe *OO*.

In both Figs. 167 and 168 the height of the shaft is adjusted by

the nut *N*, which, after adjustment, is fastened securely in such position.

In some modern direct connected plants of the vertical type the vertical bearings are placed above the generators, thus placing the entire shaft in tension (see Fig. 324, page 516).

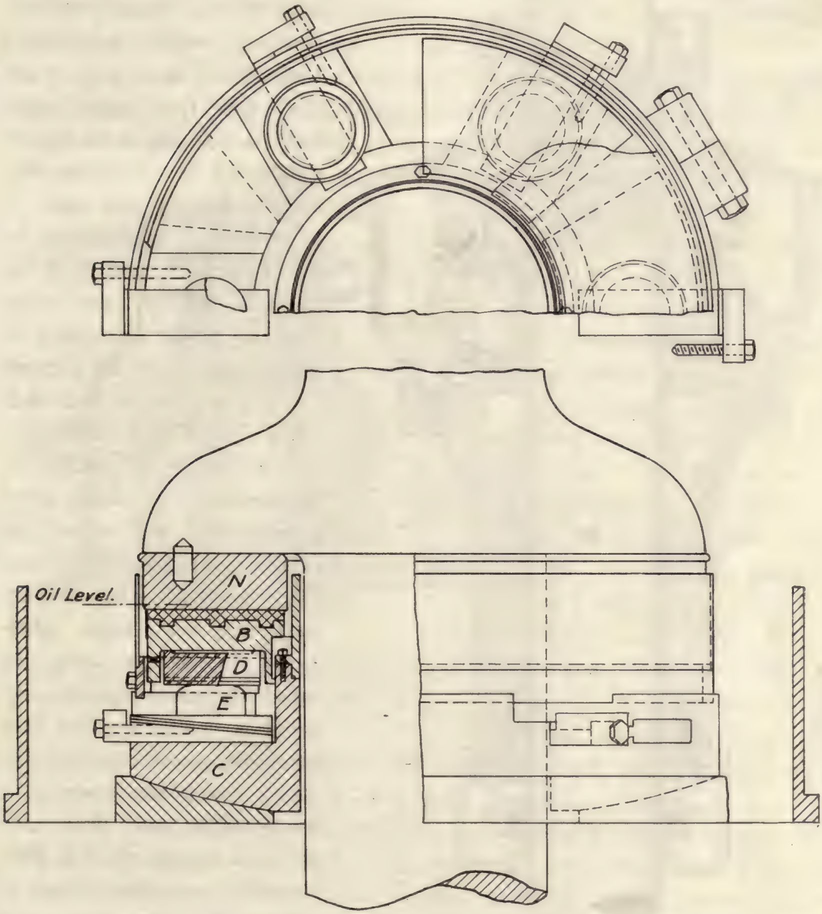


Fig. 166.—The Kingsbury Bearing (from National Elec. Light Assoc. 36th Convention, 1913). (See page 255.)

123. Horizontal Turbine Bearings.—In horizontal wheels various forms of bearing may be used according to the conditions and circumstances of their operation. When practicable the bearings should not be submerged and should otherwise be made as accessi-

ble as possible. In such cases the forms of bearings may be the same as those used on other machines subject to similar strains. In many horizontal American wheels, where submerged bearings are necessary, lignum vitae bearings are used similar in type to the upper vertical bearing before mentioned (see *T′*, Figs. 127, page 224, and 129, page 225). Such a bearing is shown in detail in Fig. 169, page 260. In this bearing the shaft *S*, is sustained in position by the blocks *TT*, which fit the recesses of the cast iron bearing block *K*, which in turn is attached to a cross tie in the case or to a pedestal *P*. The blocks *TT*, are adjusted by means of the screws *BB*, which, after adjustment are locked in position by the lock nuts *LL*. Such submerged bearings are sometimes lubricated by water only, in which case opportunity must be given for the free circulation of the water. In other cases the boxes are made tight and flow into them along the shaft is prevented by stuffing boxes at each end of the main box, the boxes being lubricated by forced grease lubrication.

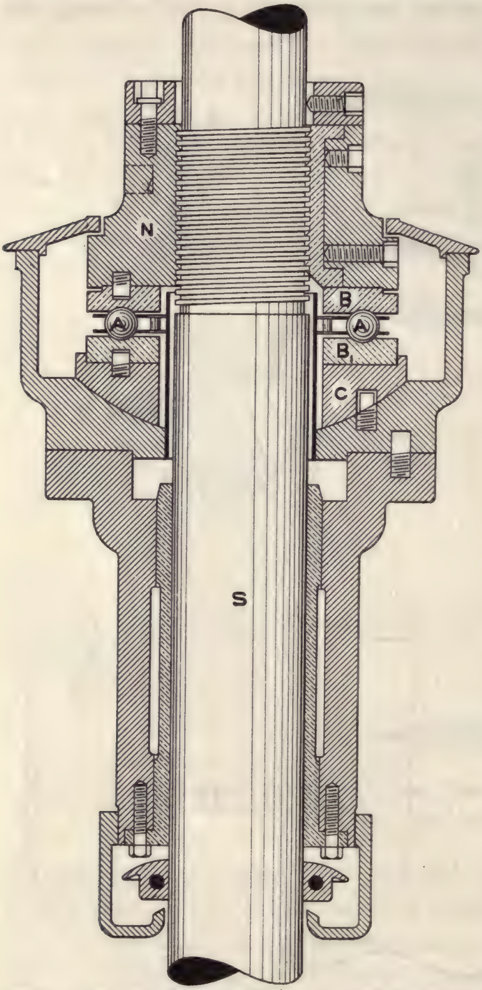


Fig. 167.—Vertical Suspension Ball Bearing (see page 256).*

Bronze boxes of the types used for other high grade machines are sometimes used for submerged bearings. In such cases great care

* Wasserkraftmaschinen von L. Quantz.

is necessary to prevent the entrance of grit-bearing waters. Such bearings are lubricated by forced oil or grease.

In forced lubrication it is desirable that both a force and return pipe be used so as to give visible evidence that the lubricant is actually reaching the bearing. In some cases bearings that would be otherwise submerged are made accessible at all times by metallic tubes (see Fig. 337, page 526) used as manholes.

124. Thrust Bearings.—Where the turbine is placed horizontally, gravity can no longer offset the thrust caused by the reaction of the turbine when the discharge is in one direction, and the thrust must therefore be overcome by the use of some form of thrust-bearing. Where other conditions permit, it is quite common practice to install two turbines on a single horizontal shaft, having their discharges in opposite directions, in which case the thrust of each turbine is overcome by the thrust of its companion (see Figs. 135, page 228, 141, page 231 and 144, page 233). In many cases, however, the arrangement, size and capacity of the wheels to be used are not such as will permit the use of twin turbines and thrust-bearing, and other means of taking up the thrust must be provided.

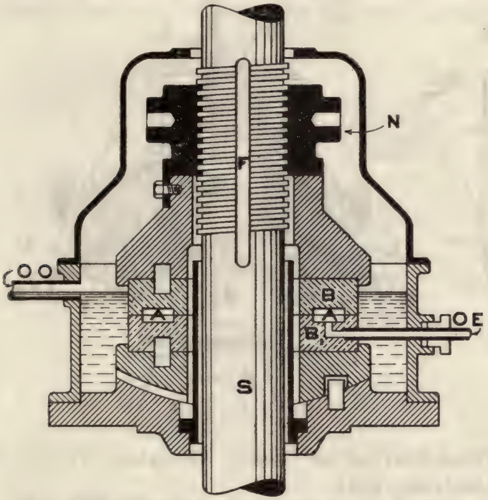


Fig. 168.—Vertical Suspension Oil Pressure Bearing (see page 256).*

In the Snoqualmie Falls turbine, manufactured by The Platt Iron Works Company (see Figs. 145, page 234, and 146, page 235), the device for taking up the thrust is thus described by the designing engineer, Mr. A. Giesler:†

"Single-wheel horizontal-shaft units are relatively infrequent in turbine practice, especially in large sizes, where the thrust of a single runner is large enough to require careful consideration. The thrust is made of two parts: (1) that due to the static pressure or

* Wasserkraftmaschinen von L. Quantz.

† See "Engineering News" of March 29, 1906.

effective head of water at the various points of the runner surface; and (2) that due to the deflection of the water from a purely radial path through the wheel. As concerns the first part, the front face of the wheel is pressed upon by a pressure varying from the supply head at the outer circumference to the discharge pressure (vacuum) at the inner edge of the vanes, which latter extends over the whole central area of the runner (and shaft extension). The rear face of the runner is subjected to the pressure of water leaking through the radial air-gap between casing and runner, substantially equal to the

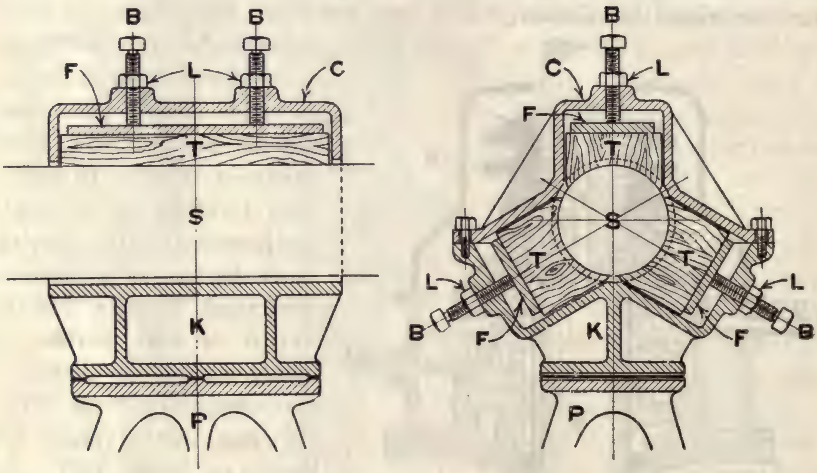


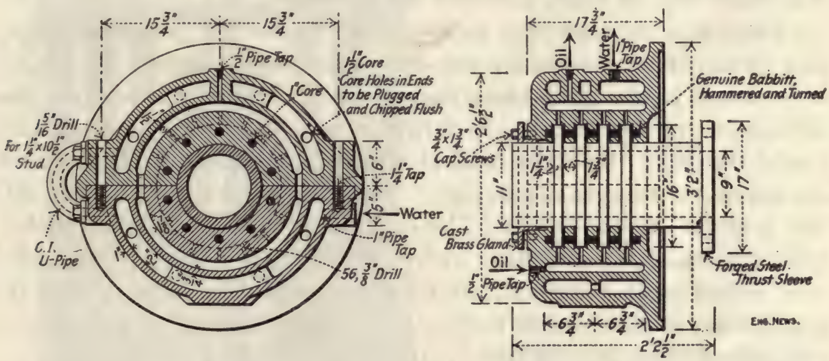
Fig. 169.—Horizontal Lignum Vitae Bearing as Used in American Turbines (see page 258).

supply head. This greatly over-balances the pressure on the front face, and the resultant thrust is to the right in Fig. 145, page 234 (toward the draft tube). The discharge ends of the vanes, being curved transversely, also have a pressure component directed toward the right. The velocity effect produces a thrust directed toward the left, but this is very small and does not materially reduce the pressure thrust.

"By far the larger part of the pressure thrust is eliminated by venting the space back of the runner into the discharge space. Six holes through the wheel near the shaft, indicated in Fig. 145, page 234, have this function. The water leaking in through the air-gap is continuously discharged through these vents into the draft-tube, and the accumulation of any large static pressure back of the wheel is thereby avoided.

"The average pressure on the front of the runner, however, is always lower, and the resultant thrust is therefore toward the draft-tube, though its amount varies considerably, being greatest for full gate opening. This thrust is taken up by the balancing piston immediately back of the rear head of the wheel case, and the ultimate balance and adjustment of position is accomplished by the collar thrust-bearing behind the balancing piston.

"The balancing piston is a forged enlargement of the shaft, finished to a diameter of seventeen inches, which works in a brass sleeve set in a hub-like projection on the back of the wheel-housing. The inside of the sleeve has six circumferential grooves, each one



Cross Section. Longitudinal Section.

Fig. 170.—Thrust-Bearing Snoqualmie Wheels.

inch wide and one-quarter inch deep, as water packing. The chamber in front of the piston communicates by a pipe (containing a strainer) with the supply casing of the water-wheel, and therefore receives the full pressure of the supply head. The chamber back of the piston is drained to the draft-tube, so as to carry off any leakage past the piston. The device thus produces a constant thrust on the piston, directed toward the left. By throttling the pressure pipe this thrust can be adjusted as desired.

"The thrust-bearing shown in Fig. 145, page 234, and in detail in Fig. 170, consists of a group of four collars on the shaft, working in a babbitted thrust-block which is bolted to the back of the wheel-housing. The collars are formed on a steel sleeve which fits over the shaft and is bolted to the rear face of the balancing piston; this makes it possible, when the collars are worn out, to renew the bearing by dismounting the thrust-block and placing a new sleeve. The

thrust-bearing is lubricated by oil immersion. An oil chamber is cored in the block and communicates by numerous oil holes with the bearing faces; a constant flow of oil is maintained by means of oil-supply and drain-pipes. Concentric with the oil chamber and outside of it a water chamber is cored in the block. Cooling water is supplied to this chamber by a pipe from the pressure side of the turbine, and drains from the top of the bearing through a drain-pipe to the draft-tube. A U-pipe attached at one side of the bearing forms connection between the water chambers of the upper and lower halves of the block. This detail avoids makings the connection by a hole through the joint face, which would allow leakage of water into the oil-space and into the bearing.

"The balancing piston is so proportioned and the pressure supply pipe is throttled to such a point as to give exact balance (i. e., with zero thrust in the thrust-bearing) at about half to five-eighths the full output of the wheel. At larger power there will be an unbalanced thrust to the right, and at smaller output to the left, which are taken by the thrust-bearings. The maximum thrust on the collars is about 25,000 pounds. The collars are two and one-half inches high (two and three-eighths inches effective) by thirteen and one-half inches mean diameter, giving a total effective bearing area on four collars of 418 square inches. The maximum collar pressure is thus about sixty pounds per square inch."

125. The Chute Case.—The chute case (see Figs. 127, page 224, 129, page 225 and 171, page 263) consists of the fixed portion of the turbine to which are attached the step and bearings of the wheel *T*, the guide passages *g* which direct the passage of the water into the turbine bucket, and the gates *G*, which control the entrance of the water, and also the case cover *C*. The case cover keeps the wheel from contact with the water except as it passes through the guide and gates. To the chute case is usually attached the apparatus and mechanism for manipulating or controlling the position and opening of the gate (*A, P, Gr,* etc.). In vertical turbines a tube *d*, is usually attached to the lower ring, forming a casing in which the lower portion of the turbine revolves and on which the bridge tree *t*, holding the step bearing is attached. When this tube is no longer than one diameter it is usually called the turbine tube; but when it is considerably extended, it is termed a draft tube.

The design of the turbine tube depends largely on the character of the wheel. Some wheels discharge inward and downward, some

almost entirely downward, some downward and outward, and in some cases the wheel discharges in all three directions. For the best results the tube should be so designed that the water from the wheel shall be received by it with no radical change of velocity and so that the remaining velocity will be gradually reduced and the water discharged at the lowest practicable velocity.

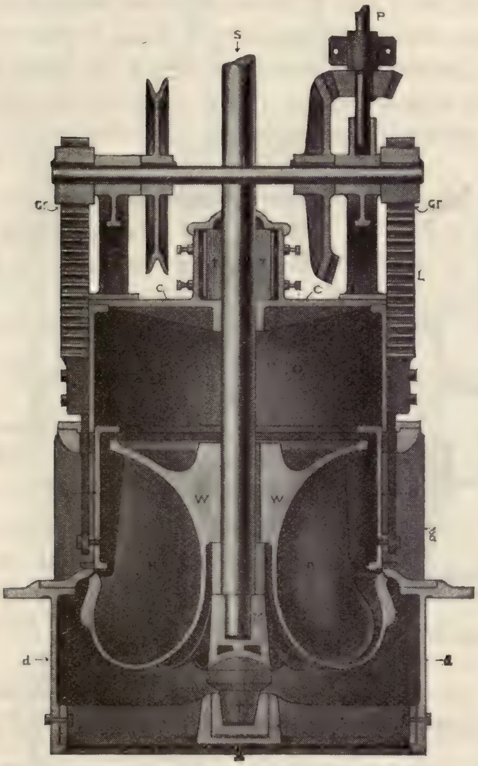


Fig. 171.—Vertical Turbine With Cylinder Gate (Wellman, Seaver, Morgan Co.). See page 262.

The chute case and its appurtenances should be so designed that the water will enter the bucket with the least possible shock or resistance at all stages of gate and with a gradual change in velocity, and will discharge from the buckets into the turbine tube with as little eddying as possible and be evenly distributed over the cross-section of the tube so as to utilize the suction action of an unbroken column of water. The case must also be designed of sufficient strength to sustain the weight of the turbine wheel and so that the step bearings are accessible and can be readily replaced or adjusted. The arrangement of the case must also be such that the openings between the wheel and the case are as small as practicable and the line of possible leakage will be as indirect as possible so as to avoid leakage loss.

Most chute cases are either cast or wrought iron. Cast iron usually lends itself to a more satisfactory design for receiving and passing the water without sudden enlargement and opportunities for losses by sharp angles and irregular passageways. Wrought iron, while not always lending itself readily to designs which elim-

inate all such losses, possesses much greater strength for a given weight which is a great advantage under some conditions.

126. Turbine Gates.—Three forms of gates are in common use for controlling the admission of water into reaction turbines. The cylinder gate consists of a cylinder closely fitting the guide that by its position admits or restricts the flow of water into the buckets. Fig. 171 is a section of a turbine of the McCormick type, manufactured by the Wellman, Seaver, Morgan Company, having a gate of this type *GG,* between the guides and runners, which is shown closed in the cut. The gate is operated by the gearing *Gr,* which raises it into the dome *O,* through connection with the governor shaft *P*. This same type of gate is used over the discharge of the Niagara-Fourneyron turbine (see *GG,* Fig. 120, page 219), and over the inlet of the Geylin-Jonval turbine *GG,* Fig. 121, page 220.

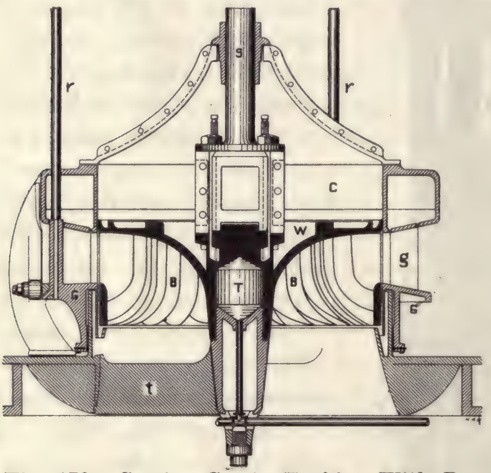


Fig. 172.—Section Swain Turbine With Drop Cylinder Gate.

A modified form of the cylinder gate is that used by the Swain Turbine Company (see Fig. 172), which is lowered into opening instead of being raised into the dome as in Fig. 171, page 263.

A similar modification, called a sleeve gate by its designer, J. W. Taylor, is shown in Fig. 173, page 265.

When partially closed the cylinder gate causes a sudden contraction in the vein of water which is again suddenly enlarged in entering the runner after opening the gate (see Fig. 176, page 268). These conditions produce eddying which results in decreased efficiency at part gate (see Figs. 172 and 173).

The wicket gate when well made, is perhaps the most satisfactory gate, especially for moderate and high heads. It can be balanced for one position only, and should usually be so designed that in case the governor mechanism should break, or become disabled, the

tendency will be to drift shut, so as to avoid overspeeding as far as possible.

These gates are illustrated by *GG*, Figs. 129 and 130, page 225, which illustrate the wicket gate of the Samson turbine of The James Leffel & Company, and Fig. 174, page 266, which shows the wicket gate of the Wellman, Seaver, Morgan Company. In both cases the wickets are connected by rods with the eccentric circle and through an arm and section with the gearing *Gr*.

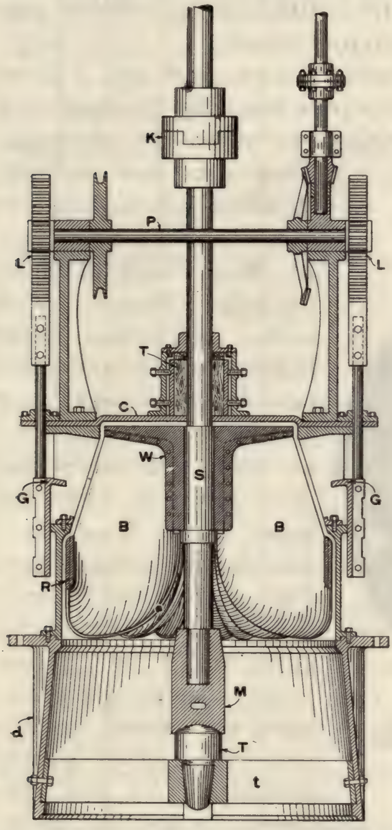


Fig. 173. — Section Showing Taylor Sleve Gate (see page 264).

Figure 127 and 128, page 224, show the wicket gate of the Improved New American, and Figs. 145, page 234, nad 146, page 235, show the wicket gates of the Snoqualmie Falls turbine, manufactured by The Platt Iron Works. In both the New American and Snoqualmie wheels, the gates are moved by a gate ring (see *Gr*, Fig. 127, page 224). Figs. 177 and 178, page 269, show the details of the wicket gates and connection of the same to the gate ring of the Snoqualmie Falls turbine.

The tendency to produce eddying is much reduced in well designed wicket gates, although the sudden enlargement of the reduced vein at part gate undoubtedly reduces the efficiency of the wheel (see Fig. 179, page 270, A and B).

The register gate (see *G*, Fig. 180, page 271) consists of a cylinder case with apertures to correspond with the apertures in the guides *g*, and is so arranged that, when in proper position, the apertures register and freely admit the water to the wheel, and is also so constructed that when properly turned the gate cuts off the passage completely or partially as desired.

Considerable eddying is produced by the partially closed register gate, with a consequent decrease in part gate efficiency (see Fig.

181, page 271). The cylinder gate is usually the cheapest and simplest form of gate, but the wicket gate, if properly designed and constructed seems to admit of the entrance of water into the bucket with least possible resistance and eddying, and in the most efficient manner. This form of gate is the most widely used in high-grade turbine construction at the present time, although the cylinder gate is largely in use and has given satisfactory results.

In the Engineering Record for April 17, 1915, page 485, is given two tests of turbines made by the Holyoke Machine Company, one being a wicket and one a cylinder gate wheel of the same diameter. These tests show as noted by Mr. H. B. Taylor (see Eng. Rec., May 29th, page 22) that except at about full power, the wheel with wicket gates has much higher efficiencies and would have shown still better comparative results had the runner used with the wicket gates given as high efficiency as the runner used with the cylinder gate (see Fig. 175, page 267).

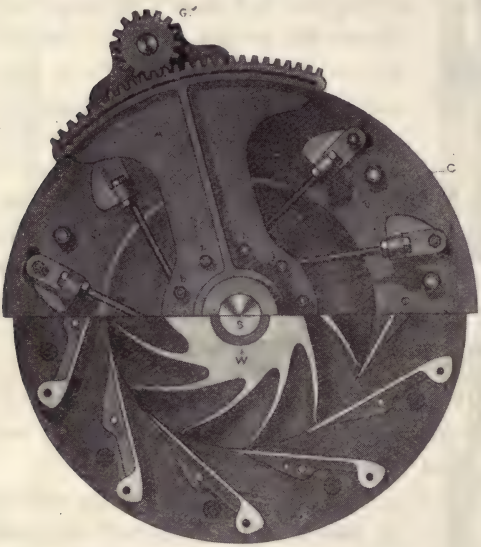


Fig. 174.—Wicket Gate of the Wellman, Seaver, Morgan Co. (see page 265).

In impulse wheels the gates are usually so arranged that the guide passages are opened one at a time instead of all opening partially, as in part gate conditions with the reaction wheel. This results in less loss in the eddying caused by part gate. Fig. 182, page 272, shows the type of gate used by The Platt Iron Works in their Girard turbines where the guide passages are arranged symmetrically in three groups about the wheel. In the tangential wheel, where a single nozzle is used, the most efficient method found for reducing the opening is with the needle as illustrated in Fig. 183, page 273. This figure shows a cross-section of the Doble needle nozzle, a form which gives a high velocity coefficient under a very wide range of opening. The character of the stream from a needle nozzle when greatly reduced is shown by Fig. 184, page 273.

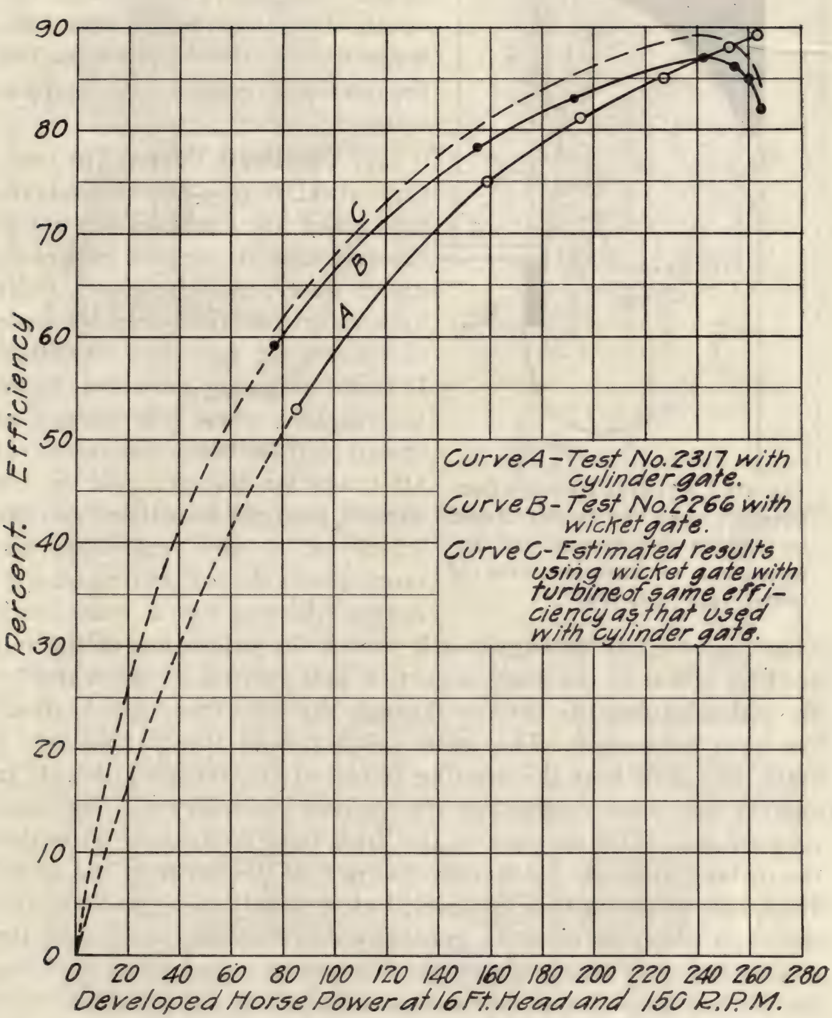


Fig. 175.—Comparative Efficiencies Obtained by Use of Cylinder and Wicket Gate (see page 266).

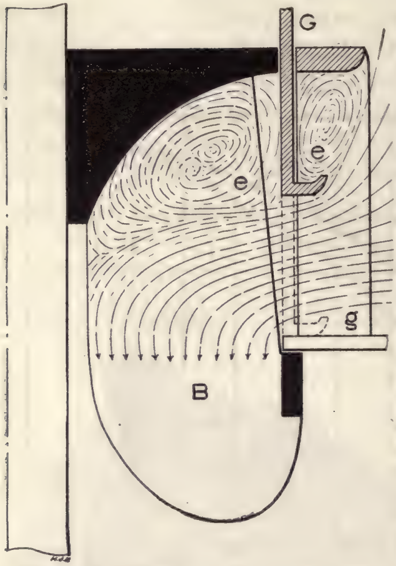


Fig. 176.—Showing Cylinder Gate Partially Open and Eddies Caused by Sudden Contraction and Enlargement of Entering Vein of water (see page 264).

where the clear and solid stream gives evidence of high efficiency. If the flow of water through the nozzle is regulated by throttling the water with a valve before it reaches the nozzle, a very low efficiency results.

127. The Draft Tube.—The reaction wheel is of particular advantage under low heads on account of the fact that it can run efficiently under water, and therefore, under backwater conditions, can be made to utilize the full head available. It is not necessary, however, to set the reaction wheel low enough so that it will be below tail water at all times for the principle of the suction pipe can be utilized and the wheel set at any reasonable distance above the tail water and connected thereto by a draft tube which, if properly arranged, will permit the utilization of the full head by action of the draft or suction pull exerted on the wheel by the water leaving the turbine through the tube from which all air has been exhausted. The water issuing from the turbine into a draft tube, which at the starting is full of air, takes up the air in passing and soon establishes the vaccum necessary for the draft tube effects. The function of the draft tube is not only to enable the turbine to utilize by suction that part of the fall from the wheel discharge to the tail water level, but it should also gradually increase in diameter so as to gradually decrease the velocity of the water after it is discharged from the turbine wheel, thus enabling the turbine to utilize as much as possible of the velocity head with which the water leaves the turbine.

In the modern turbine of high specific power operated under low heads, the velocity head to be recovered in the draft tube sometimes amounts to almost one-third of the total head, hence high turbine efficiencies are only possible with practically perfect draft tube conditions, as a vacuum more or less complete will be established in

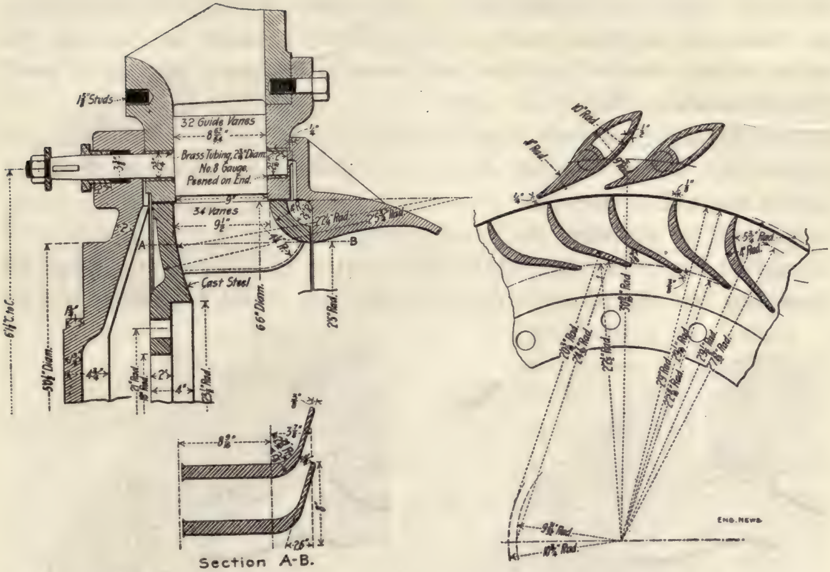


Fig. 177.—Showing Relations of Gate Guides and Buckets in Snoqualmie Falls Turbine. Platt Iron Works Company (see page 265).

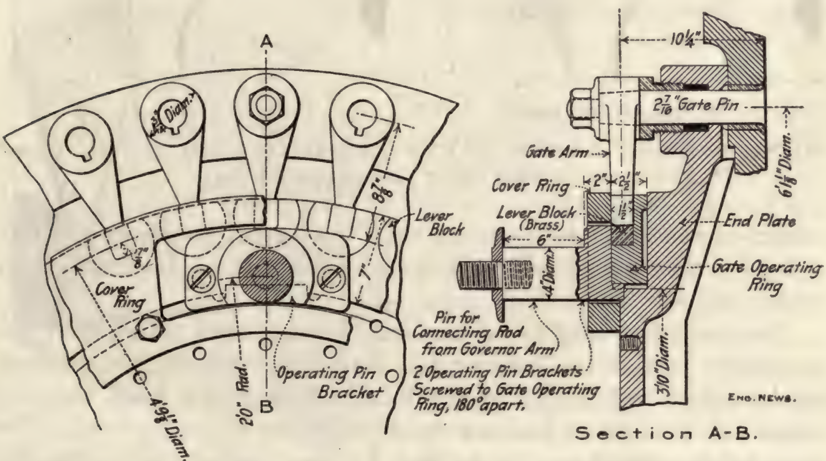
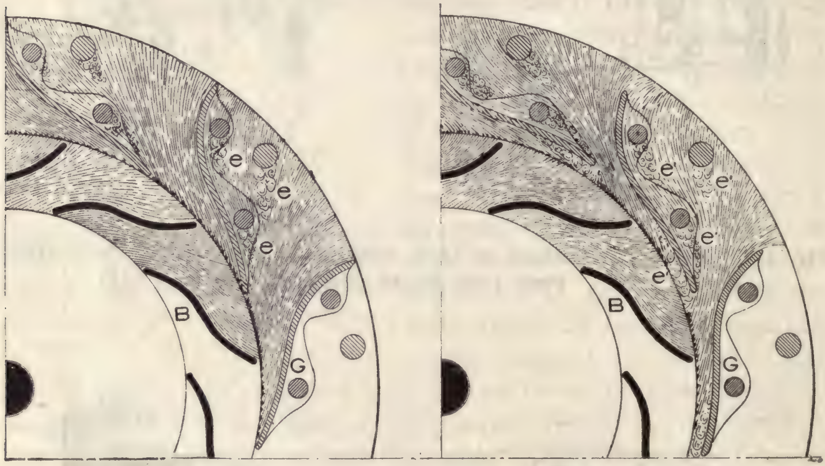


Fig. 178.—Showing Rigging for the Operation of Wicket Gate in Snoqualmie Falls Turbine. Platt Iron Works Company (see page 265).

the draft tube and, therefore, the draft tube must be strong enough to stand the exterior pressure due to the vacuum so created. In order to perform its functions in a more satisfactory manner, it must also be made perfectly air tight.

One of the great advantages in the use of the draft tube is the possibility, by its use, of setting the wheel at such an elevation above the tail water that the wheel and its parts can be properly inspected, by draining the water from the wheel pit. Otherwise it would be necessary to install gates in the tail race and pumps for pumping out the pit in order to make the wheel accessible. The-



A. Gate Wide Open. B. Partial Gate.

Fig. 179.—Showing Condition of Flow Through Open and Partially Closed Wicket Gates (see page 265).

oretically, the draft tube can be used of as great length as the suction pipe of a pump, and this is probably true of draft tubes for very small wheels.

Practically the height of the wheel above the tail water must not exceed the sum of the velocity and draft heads and the temperature of the water and height of the location above sea level must be taken into account as limiting the possibilities of the vaccum. The total height should usually be less, by four feet or more, than the barometric limits of a perfect vacuum. Should the vacuum be broken, irregular running will result and corrosion is apt to occur on account of the consequent liberation of oxygen.

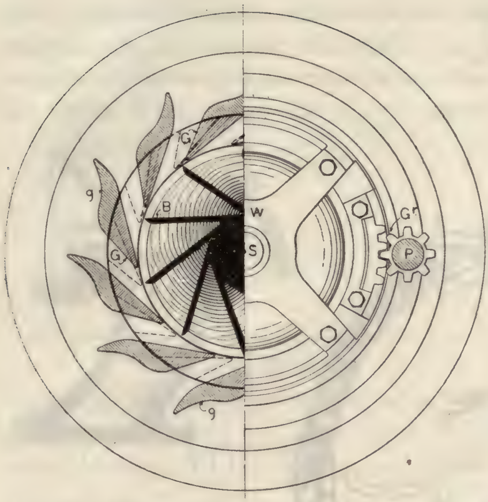


Fig. 180.—Register Gate. Platt Iron Works Company (see page 265).

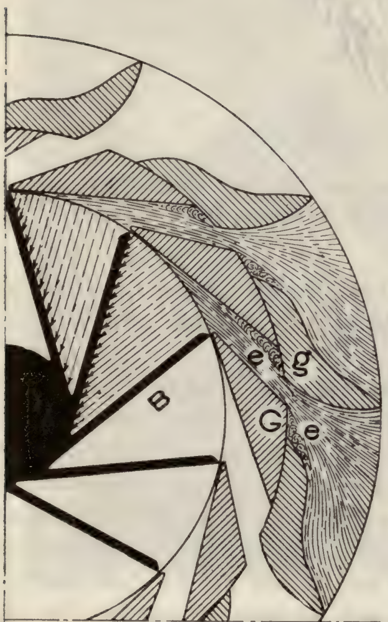


Fig. 181.—Showing Eddying Caused by Partial Closure of Register Gate (see pages 265 and 266).

The draft tube should be as short as practicable for large wheels, for its success in the utilization of the head depends on the maintenance of a solid, unbroken column of water, which is difficult to maintain in large tubes. As the size of the wheel increases the difficulties of maintaining a vacuum increase and the length of the draft tube should correspondingly decrease. It is practically impossible to maintain a working head with large turbines through long draft tubes with the turbine set at great distances above the water. Long draft tubes should, as a rule, be avoided, and in all cases where draft tubes are used, they should be as straight and direct and as nearly vertical as practicable. It is the principle of the draft tube that permits horizontal shaft wheels to be utilized, as otherwise, with this type of machinery, only a small portion of the head could be used to advantage under normal conditions, for such wheels being often direct connected to the machinery are, of necessity, placed above the tail water. The draft tube is commonly of iron or steel in the ordinary small installation but in modern high grade installation they are commonly molded directly in the concrete of the station or wheel foundation.

On the Fourneyron turbine Boyden used what he termed a diffuser (see Fig. 185, page 274). The main purpose of the diffuser, and of the conical tube as well, is to furnish a gradually enlarged passage through which the velocity of the water as it leaves the

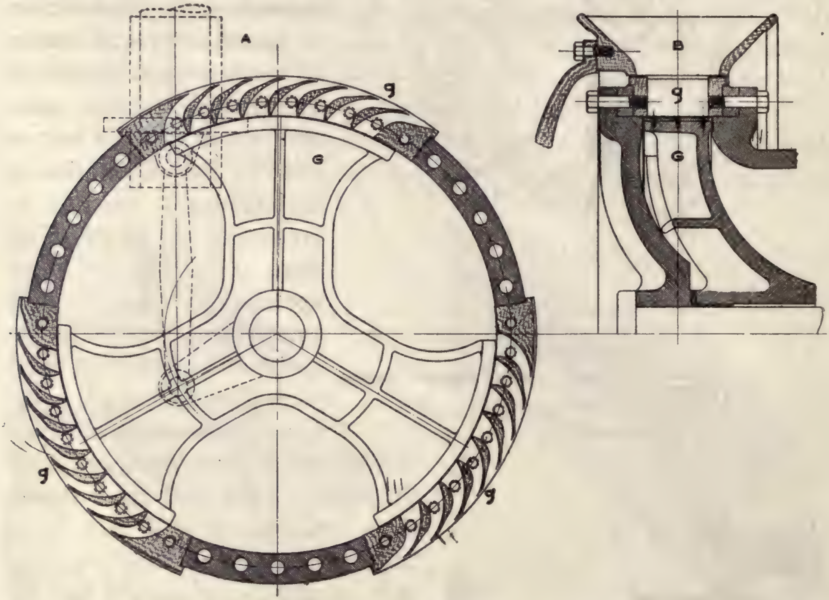


Fig. 182.—Gates and Guides of Girard Impulse Turbine. Turbine Design as Modified for Close Speed Regulation, G. A. Buvinger, Proc. Am. Soc. M. E., Vol. XXVII (see page 266).

wheel is so gradually reduced as to enable the velocity head to be utilized in the wheel, thus saving head which would otherwise be lost.

It has already been noted that impulse wheels of the Pelton and Girard types cannot operate satisfactorily submerged, and must be set at such positions that they will be above the tail water at all times. In many localities where the variation in the surface of tail waters is considerable, this means a large relative loss in the head utilized, and this type of wheel will therefore not be practicable except under high head conditions and where the loss entailed by the rise and fall of the tail water will be inconsiderable. An attempt has been made, however, to so design a draft tube that a vacuum will be established and maintained below the wheel, in

* From Bulletin No. 6, Abner Doble Co.

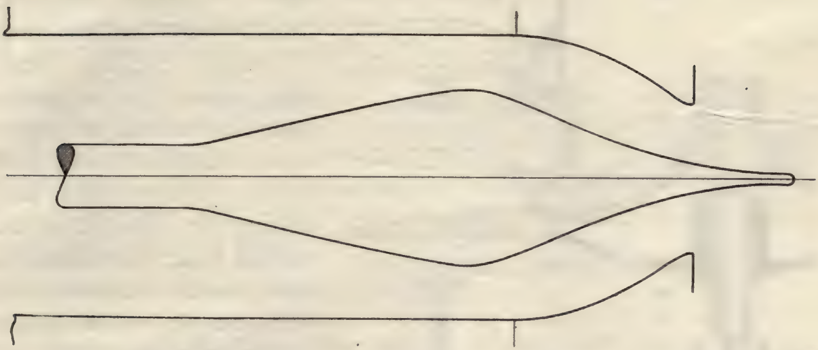


Fig. 183.—Cross-section of Doble Needle Nozzle (see page 266).*

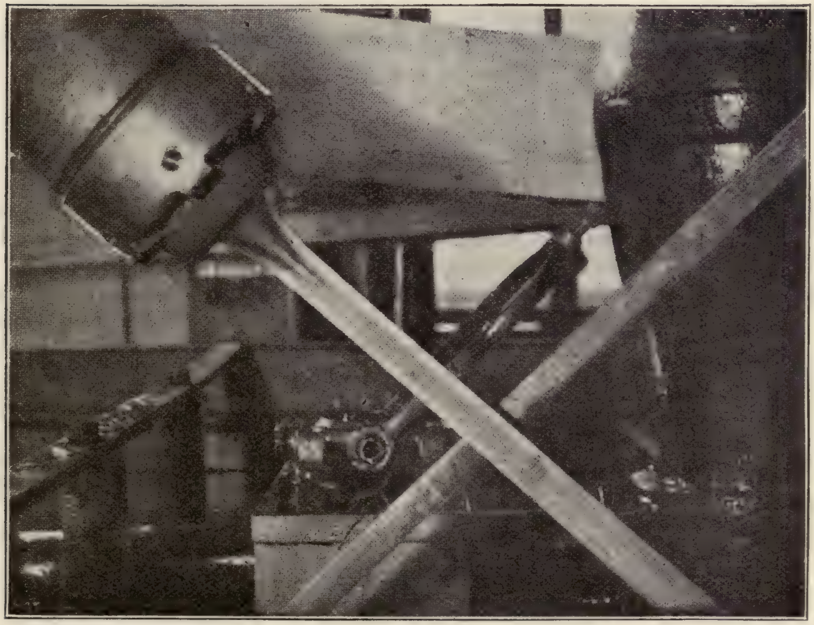


Fig. 184.—Stream From Doble Needle Nozzle (see page 266).*

* From Bulletin No. 6, Abner Doble Co.

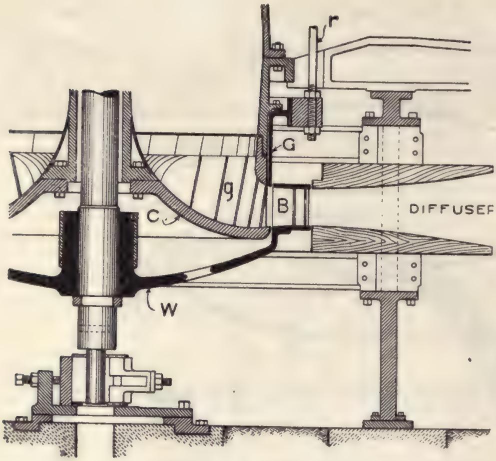


Fig. 185.—Boyden Diffuser Used With Fourneyron Turbine (see page 272).

such a manner, however, that the water will not come in contact with the wheel. The vacuum is so maintained as to hold the water at an established point just below the wheel, thus permitting the wheel to utilize the full head except for the small clearance between the wheel and the water surface in the draft tube. This arrangement is shown in Figs. 151, page 240 and 154, page 242, as applied by The Platt Iron Works Company to a Girard turbine.

The successful application of this design is understood to be very difficult and it is not in general use with impulse wheels.

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CHAPTER XI

HYDRAULICS OF THE TURBINE

128. Symbols Used in Chapter.—In the discussion of this chapter the letters and symbols used have the following significance:

a = Area of gate orifice or orifices.
α = Angle of deflection of jet.
β = Supplement to angle of deflection $= 180° - \alpha$.
D, D_d = Diameters of homologous wheels.
D_1, D_{48} = Diameter of wheels of one inch and forty-eight inches, respectively.
E = Energy in foot pounds per second.
e = Efficiency; when used as a subscript to any coefficient, it indicates that the corresponding value is for the condition of maximum efficiency of the wheel.
F = Force producing pressure or motion.
g = Acceleration of gravity.
h, h_h = Effective heads under which wheels are to operate.
h_1, h_{12} = Effective heads of one and twelve feet, respectively.
n = r. p. m. = Revolutions per minute.
n, n_d = Revolutions per minute of homologous wheels of diameters D and D_d, respectively.
n, n_h = Revolutions per minute of homologous wheels of equal diameters under heads h and h_h, respectively.
n_1, n_{12} = Revolutions of homologous wheels of equal diameters under heads of one and twelve feet, respectively.
π = Ratio of circumference to diameter = 3.1416.
P = Horse power of turbine at any given head.
P_t = Theoretical horse power.
P, P_d = Horse power of homologous wheels of diameters D and D_d, respectively.
P, P_h = Horse power of homologous wheels of equal diameter under heads h h_h, respectively.
P_1, P_{12} = Horse power of homologous wheels of equal diameters under heads of one and twelve feet, respectively.
q = Discharge in cubic feet per second.
q, q_d = Discharge in cubic feet per second of homologous wheels of diameters D and D_d, respectively.
q_1, q_h = Discharge in cubic feet per second of homologous wheels of equal diameters under heads of h and h_h, respectively.
q_1, q_{12} = Discharge in cubic feet per second of homologous wheels of equal diameters under heads of one and twelve feet, respectively.
r_1 = Internal radius of wheel.
r_2 = External radius of wheel.

S = Space passed through by force acting.
u' = Velocity of periphery of runner at discharge.
u'' = Absolute velocity of water leaving wheel.
u_r = Velocity of water at discharge relative to velocity of wheel.

v = Theoretical spouting velocity of water in feet per second = $\sqrt{2gh}$.
v_a = Average velocity.
v'' = Velocity of water entering the runner.
v' = Velocity of periphery of runner at entrance.
v_r = Velocity of entering water relative to velocity of wheel.
W = Total weight per second.
w = Weight of unit of water = 62.5 lbs.
X = When used as a subscript denotes that the values of D, n and h thus indicated are directly related and vary with each other in order to fulfill the requirements of the formulas.

TURBINE CONSTANTS

ϕ = Ratio of peripheral velocity of wheel to spouting velocity of water = $\frac{v'}{v}$
c = Coefficient of discharge through area or areas a.
c', c'', etc. = Various modifications of the coefficient of discharge.
$\triangle$ = Speed coefficient = speed or r. p. m. of a one inch wheel under one foot head.
K = Discharge coefficient = discharge of a one inch wheel under one foot head.
p = Power coefficient = power of a one inch wheel under one foot head.
$\mathfrak{P}$ = Specific power coefficient = power of a wheel at one revolution per minute under one foot head.
N_u = Unity speed or type characteristic (sometimes called specific speed) = the revolutions of a wheel of such diameter that it will give one horse power under one foot head.
N_s = Specific speed as adapted to the metric system.

129. **First Principles.**—Shocks caused by sudden changes in the direction of flow or by sudden reduction in the area of passages, and sudden enlargements resulting in the formation of eddies in the water during its passage through hydraulic machinery, are always sources of loss of head and consequent loss of energy. *To secure the highest efficiency a water wheel must be so designed that the water will enter the wheel without shock, and leave it without velocity,* so far as practicable. This should be understood as meaning that a water wheel must be so designed, constructed and operated that the water in its approach to the buckets of the wheel, in its entrance into and in its passage through the buckets, should be so guided as to prevent losses from sudden contraction or enlargement or abrupt changes in direction of flow, and under such conditions, that by the

gradual reduction of its absolute velocity, the water will transfer its energy to the bucket so that the water will leave the bucket without absolute velocity, and consequently without energy, as nearly as practicable.

To prevent shock, the water must enter the buckets of any water wheel in a direction tangential to the curved surface of the bucket at that point and must transfer its energy by weight, pressure or impulse as it gradually changes its direction of flow relative to the direction of motion of the bucket (see Section 131, page 281, and Section 134, page 285). A comparison of the effect of the impact due to the jet striking the bucket in other than a tangential direction is shown by a study of Equations (70), (71) and (72), page 281. In all cases, the energy remaining in the water as it leaves the wheel, either in the form of velocity or head, is entirely lost unless partially recovered by the use of a draft tube or other similar device.

The greater the reduction in the velocity of the water leaving the wheel, or draft tube where draft tube is used, the greater the proportion of energy that can be utilized; but there comes a limit beyond which it is not practicable to reduce velocities. This limit varies with different conditions, and may and usually will result from limiting the expense in the building of raceways or in the construction of the machine itself. The theoretical limit of reduction in velocity is reached when the energy lost equals the interest and depreciation on the works necessary for saving it. A point might also be reached where the friction expended in the large machine needed to further reduce the velocity of the discharging water, would consume more energy than would be lost in rejecting the discharge at a higher velocity. In practice it is found that about two or three feet per second are satisfactory velocities at which to reject or discharge the water used by water wheels. These velocities represent heads of from .062 to .14 feet, or from three-quarters to slightly less than two inches. The limit of the velocity of discharge should, however, be fixed for each individual case and after all conditions have been fully considered.

130. The Transformation of the Energy of Water.—The energy of water may be exerted and transformed into mechanical energy in water motors by weight, by pressure, by reaction and by impulse. In most motors there is a combination of these transformation methods by which the energy is exerted and the transfer takes place.

In the gravity wheel (Fig. 113, page 205) weight is perhaps the most important means of transformation, but the impulse or pressure produced by the entering water is not without effect.

In the reciprocating hydraulic motor, *pressure* is the most important method of transformation, but certain velocity changes are also active in the transfer.

When the entire dynamic energy of water is converted into kinetic energy through velocity due to its head, the energy of the issuing streams may be transformed by reaction on the vessel of which the orifice is a part and may produce motion in that vessel, if it be free to move, or it may produce motion in another body by impulse through the extinction of the momentum of the jet in impinging against it.

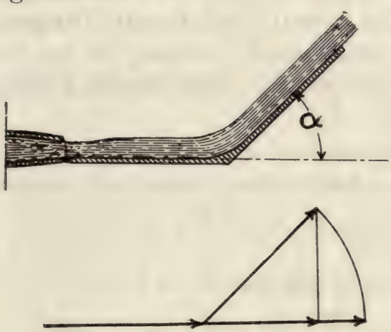


Fig. 186.

Fig. 187.

Fig. 188.

Action of the Jet Defected in Various Ways.

These equal and opposite forces are shown:

1. By the force required to sustain a hose nozzle against the reaction of a fire stream, and

2. By the force of the jet, from the nozzle so sustained when exerted against any object in its course.

In Barker's Mill (Fig. 8, page 7) reaction is the principal means of energy transformation, but the pressure along the curved channels is also an element of transfer. If the flat plate in Fig. 187, is one of a series of similar buckets on a tangential wheel as in Fig. 6, page 5, the transfer of energy may be regarded as entirely by impulse. If, however, the buckets be of the type shown by Fig. 16, page 17, the transfer of energy is partially by impulse and partially by the reactive pressure produced by the change in velocity and direction of the water jet.

When, as in most so-called impulse and reaction wheels of modern design, the construction and speed are such that the jet or jets of water enter the buckets tangential to the curved surfaces, they deliver their energy by changing the kinetic energy of the water jets to pressure energy as the paths of the jets are changed and their velocities checked. The transfer of energy in the so-called impulse wheel cannot be regarded as due to impulse as above defined but occurs through reactive pressure produced by the gradual extinction of absolute velocity. In the so-called reaction turbine, the transformation of energy is most largely through pressure exerted throughout the breadth of the bucket and producéd in an identical manner to that in the impulse wheel. In this case the buckets are filled and as the cross-section is gradually reduced towards their outlets the pressure changes with the velocity and, besides the pressure due to a change in direction and absolute velocity of the jets of water, there is perhaps an element of the purely reactive effect of the Barker's Mill.

From elementary principles of hydraulics, it is established that a jet of water spouting freely from a frictionless orifice will acquire a velocity

(42) $$v = \sqrt{2gh}$$

and will possess energy in foot pounds per second as follows:

(4) $$E = \frac{Wv^2}{2g} = \frac{qwv^2}{2g}$$

The energy of the jet leaving an orifice is the product of a force F, which acts on a weight of water qw, through a space S, in one second of time, giving the velocity v. This energy measured in foot pounds is

(65) $$E = FS$$

The space passed through by the force in one second in raising the velocity from o to v is

(66) $$S = v_a t = \frac{v}{2}$$

Hence, from equation (3),

(67) $$E = \frac{Fv}{2}$$

From equations (4) and (67) therefore

(68) $$\frac{Fv}{2} = \frac{qwv^2}{2g}$$

As $q = av$ it therefore follows that

$$F = \frac{qwv}{g} = \frac{awv^2}{g} = 2awh \tag{69}$$

The force F, which may be exerted by a jet impinging against a surface depends on the momentum of the moving stream of water and is directly proportional to its velocity. It is also a function of the angle through which the jet is deflected. If friction be ignored, the stream will be deflected without change in velocity, and the component of the force exerted against the surface in the original direction of the jet will be equal to the momentum of the original stream less the component, in the original direction, of the momentum of the diverted jet (see Fig. 186).

$$F = \frac{qwv}{g} - \frac{qwv}{g}\cos\alpha = \frac{qwv}{g}(1 - \cos\alpha) \tag{70}$$

If the jet impinges against a flat surface (see Fig. 187)

$$\alpha = 90°, \text{ Cos } \alpha = 0 \text{ and}$$

$$F = \frac{qwv}{g} \tag{71}$$

If the jet is deflected 180 degrees by means of a semi-circular bucket (see Fig. 188)

$$\text{Cos } 180° = -1, \text{ and therefore}$$

$$F = 2\frac{qwv}{g} \tag{72}$$

131. The Impulse Wheel.—Impulse water wheels utilize the kinetic energy of a jet entering the buckets attached to the circumference of the wheel, tangentially to their curved surfaces. The bucket must move under the impulse in order to transform the energy of the water into work. Let the ratio of v', the velocity of the periphery of wheel, to the velocity v of the jet be indicated by ϕ then

$$\phi = \frac{v'}{v} = \frac{v'}{\sqrt{2gh}} \text{ and } v' = \phi v \tag{73}$$

In determining the force F, exerted upon the moving bucket, the relative instead of the actual velocity of the jet must be considered and it is readily seen that the value of the relative velocity v_r will be as follows:

$$v_r = v - \phi v = (1 - \phi)v \tag{74}$$

The relative weight of water that strikes a single bucket per second will also be less on account of the movement of the buckets,

but as new buckets constantly intercept the path of the jet the total amount of water effective is equal to the total discharge of the jet. Hence from equations (70) and (74)

$$(75) \qquad F = (1 - \cos \alpha) \frac{qwv}{g} (1 - \phi)$$

The work done upon the buckets per second is equal to the force *F,* times the distance ϕv through which it acts, i. e.

$$(76) \qquad E = F \phi v = (1 - \cos \alpha)\ (1 - \phi) \frac{qwv}{g} \phi v$$

This is a maximum when $(1 - \phi)\ \phi$ is a maximum the solution of which gives $\phi = .5$

Substituting $\phi = .5$ and $\alpha = 180°$, in equation (76), there is obtained

$$(77) \qquad E = \frac{qwv^2}{2g}$$

That is, *E* equals the entire energy of the jet (see equation 4), and hence the theoretical efficiency when $\phi = 0.5$ is 100 per cent.

Another criterion for maximum efficiency is that the absolute velocity of the water in leaving the bucket must be zero.

When $\alpha = 180°$, the absolute velocity with which the water leaves the bucket is evidently the velocity relative to the bucket minus the velocity of the bucket or

$$(78) \qquad u'' = (1 - \phi)\ v - \phi v = v - 2\phi v = 0$$

This gives

$$(79) \qquad \phi = 0.5$$

132. Illustration of Flow.—Fig. 189, page 283, illustrates graphically the flow of water into and through the bucket of a tangential wheel at the most economical relative velocity. The bucket is double, each half being essentially semi-circular in section. v is the absolute velocity of the jet; v' is the absolute velocity of the bucket: v_r is the relative velocity of the jet in relation to the bucket: or

$$(74) \qquad v_r = (1 - \phi)\ v$$

The bucket is moving in the direction *BB′* and occupies successively the positions indicated by the vertical lines a, a_1, a_2, etc., in equal intervals of time. The water moves along the surface of the bucket with a uniform velocity v_r, passing successively through equal distances, indicated on the surface of the bucket by the lines b, b_1, b_2, etc., in equal intervals of time.

At each of these successive points, the jet has changed its direc-

tion and its absolute velocity. The force exerted by each particle of water in moving around the semi-circular bucket is constant in amount and normal in direction, to the surface of the bucket. Its

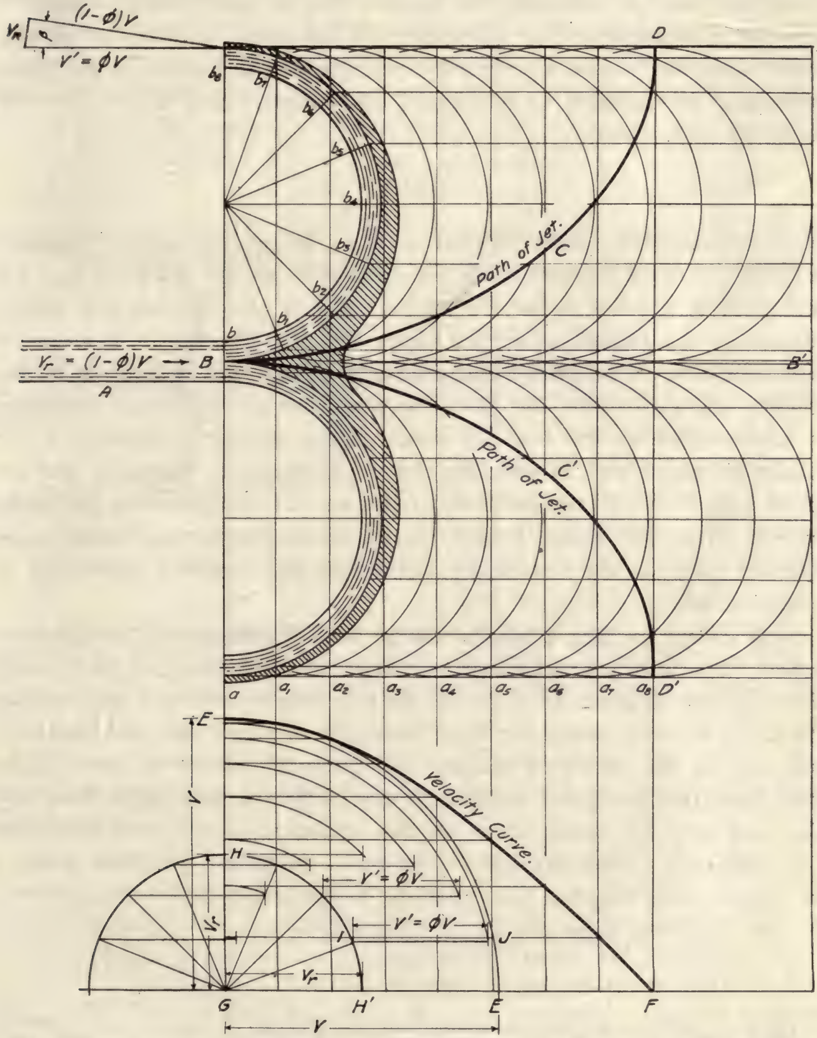


Fig. 189.—Graphical Illustration of Flow Through a Tangential Wheel Bucket (see pages 282, 285).

useful force is however the component exerted in the direction of motion. Its path through space is represented by the line *BCD*. The change in velocity is represented by the absolute velocity curve

EF, in which ordinates are the resultants obtained by applying the principle of triangle of velocities to corresponding velocities of the bucket and of the water relative to the bucket.

At the time of entering the bucket, the stream has the absolute velocity v, represented by the length of the lines GE and GE' in the lower diagram, while its velocity relative to the bucket is constant and equal to v_r, equal to the length of lines GH and GH'. For the most efficient speed,

$$v_r = \frac{v}{2} \tag{80}$$

At the end of the first interval of time, the jet has moved from the original point of contact with the bucket b, to the position b_1. Its direction and velocity in the upper half of the bucket are represented by the radius GI in the lower velocity diagram. $v_1 = \phi v$, is constant both in magnitude and direction, and this is laid off in the lower diagram on the line IJ. The resultant of these two velocities is represented by the line GJ which is the absolute velocity of the water in space, and to indicate the velocity of the water at this instant is laid off for the purpose of the velocity diagram on the ordinate a_1, from the axis GF as o in the lower diagram. In the same manner each of the remaining points on the velocity curve EF is constructed.

133. **Effect of Angle of Discharge on Efficiency.**—In an impulse wheel it is not practicable to change the direction of the water through 180 degrees as it would then interfere with the succeeding bucket. α must hence be less than 180 degrees and the absolute velocity of the water in leaving the buckets cannot be zero. The loss from this source is small as α may differ considerably from 180 degrees without much effect on the bucket pressure and hence on the efficiency. For example,—the ratio of actual pressure when α is less than 180 degrees to maximum possible pressure with $\alpha = 180°$ is (see Fig. 190, page 285).

$$\frac{(1 - \cos\alpha)\, qw\, (1 - \phi)\, \dfrac{(1-\phi)\, v}{g}}{(1 - \cos 180°)\, qw\, (1 - \phi)\, \dfrac{(1-\phi)\, v}{g}} = \frac{1 - \cos\alpha}{2} = \frac{1 + \cos\beta}{2} \tag{81}$$

$$\text{If } \beta = 8°,\ \alpha = 172°, \text{ and } \frac{1 - \cos\alpha}{2} = .995$$

showing only 0.5 per cent. reduction. The effect on the efficiency is in the same ratio.

In Fig. 189, page 283, the jet leaves the bucket as shown with a velocity, relative to the bucket, of $(1 - \phi)\ v$. If this velocity is combined graphically with the velocity of the bucket ϕv, the true absolute residual velocity v_r, of the water will be obtained. The efficiency is evidently maximum when ϕ has a value which makes v_r equal zero. When this condition exists, then

(82 $\phi v = (1 - \phi)\ v$ which gives

(79) $\phi = 0.5$

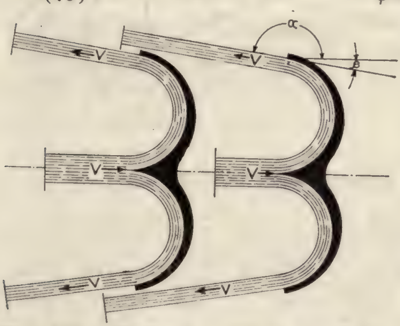


Fig. 190.—Deflection of the Jet in a Tangential Wheel Bucket (see page 285).

as obtained by two other methods, and here shown to be independent of β. This is also shown by equation (76).

The theoretical considerations thus far discussed are modified by the frictional resistance which the bucket offers to the flow of water over its surface and by the spreading of the original jet from its semi-circular section to a wide thin layer in leaving the bucket.

Further loss no doubt takes place as a result of the fact that the bucket is in its assumed position at right angles to the direction of the jet only at one instant during its rotation. Upon entering and leaving the jet it is inclined considerably to this direction and doubtless operates less efficiently. These conditions result in a much greater drop in efficiency than the above analysis would seem to indicate.

134. **Reaction Wheel.**—The flow of water through the buckets of a reaction wheel is less easily analyzed than in the case of the impulse wheel. The chief difference in the two types of wheels arises from the fact that the reaction wheel is "filled" and hence the velocity of the water relative to the buckets at any point does not remain constant but varies inversely as the cross-sectional area of the passageway and with a consequent variation in static pressure.

The path described by a particle of water in passing through the wheel has been investigated by Francis,* by a method based upon

* See "Lowell Hydraulic Experiments," p. 39.

the assumption that "every particle of water contained in the wheel, situated at the same distance from the axis, moves in the same direction relative to the radius and with the same velocity." This assumption becomes more accurate as the number of buckets increases.

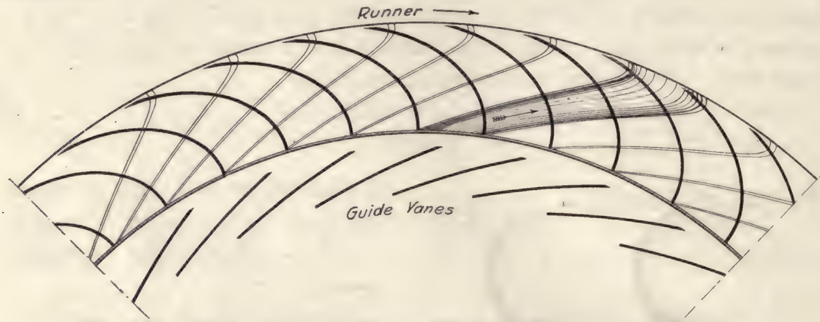


Fig. 191.—Path of Jet of Water Through Fourneyron Turbine.

Figure 191 shows the path, resulting from the application of this assumption, of the water through the "Tremont" Fourneyron wheel and Fig. 192 through the center vent wheel at the Boott Cotton Mills. The former indicates, since the jet of water is carried forward in the direction of rotation, that the water resists the rotation of the wheel until nearly to the circumference when it is suddenly deflected and leaves the wheel, as it should, in a direction nearly normal to the wheel.

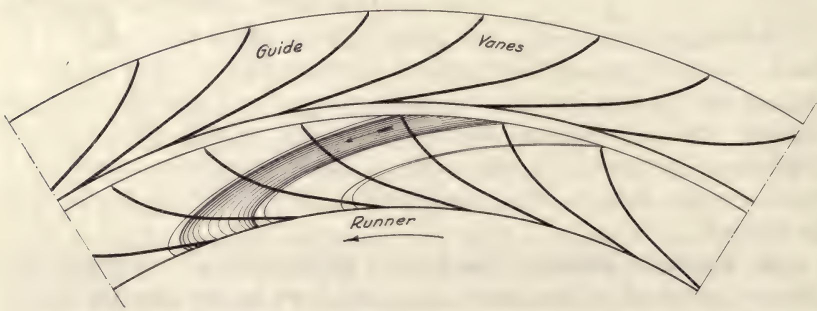


Fig. 192.—Path of Jet of Water Through Francis Turbine.

The jet of water in the Boott wheel (Fig. 192), on the other hand, shows a continual backward deflection of its path from the point where it leaves the guides, and hence a continual delivery of its energy to the wheel. This seems to indicate a more logical

condition and a better shaped bucket than that of the Fourneyron. It will be noted that the actual path of the water in this case is somewhat similar to that in the impulse wheel shown in Fig. 189, page 283.

It is well to note again that the difference in the manner of transferring energy in the so-called impulse and reaction wheels is more imaginary than real. In both cases the force exerted against the bucket of the wheel is due principally to the reactive pressure of the water caused by a change in direction as it passes around the curve of the bucket, and the consequent extinction of the absolute velocity of the water in the wheel.

135. Principles of Economy in the Design of Reaction Turbines.— For the economical operation of the reaction wheel the following principles must be observed:

First: In order that the jet of water may enter the wheel without shock, the velocity of the water as it leaves the guides and its direction relative to the velocity of the periphery of the runner and the curves of the bucket, must be tangential to the bucket blades at this point, and have a magnitude which will produce the required discharge through the cross-sectional area of the passageway.

Second: The relative velocity of the bucket and of the water relative to the bucket at the point of discharge must be such that the water leaves the bucket with the minimum practicable absolute velocity.

Third: Such residual velocity as may remain in the discharging water must be conserved and utilized so far as practicable by the proper arrangement of the draft tube.

Fourth: In all wheels it is also essential by proper design to reduce losses from friction, leakage, eddying, etc., as completely as possible.

The first principle is illustrated in Figs. 193, 194 and 195, page 289. These diagrams show the relative design of low speed, medium speed and high speed inward flow turbines in which water is delivered through certain guides AC to certain buckets AB of a turbine runner. The velocity v'', of the water entering the wheel is taken as the same in each case, but the velocities of the runner v', is different in each case. In each case the guides AC direct the water onto the buckets AB in a direction and with an absolute velocity indicated by the arrow marked v''. The velocity of the runner at the point A, where the water enters the bucket, is indicated in

amount and direction by the arrow marked v'. Combining the two velocities graphically shows the direction and velocity of the jet relative to the direction and velocity of the moving bucket at the point of entrance, as indicated by the arrow marked v_r. In order that there shall be no shock, the curve of the bucket must be tangent to the relative velocity line v_r and must have the value

$$v_r = \frac{q}{a} \tag{83}$$

in which

q = Discharge through the passageway of the bucket in cubic feet per second, and

a = Area of cross-section of the passageway, at the point of entrance *A*, in square feet measured perpendicular to the vanes.

It is apparent that only a single filament of water can follow the curve of the guide and buckets and that in the actual wheel the requirements above outlined can be only approximate.

The effect of part gate conditions upon the first principle depends upon the type of speed gate and may best be studied from Figs. 176, page 268, 179, page 270 and 181, page 271. A change in either direction or magnitude of v'' will change v_r, unless the two effects tend to neutralize, which may happen in some instances. In reaction wheels the velocity of inflow v'', through the guides may be changed by a change in the gate, while the velocity v', of the wheel will remain unchanged under normal operating condition. Under these conditions v_r will therefore change, and a change in either its direction or magnitude will produce an impact, while a decrease in the quantity of flow will result in a sudden enlargement as the water enters the runner, and therefore a loss.

The wicket gate, when properly designed, has given rise to part gate efficiencies more nearly approaching those of impulse wheels than with gates of any other type (see Fig. 175, page 267).

The second principle of economy, that of minimum residual velocity of the water in leaving the buckets, is also shown graphically in Figs. 193, 194 and 195, page 289, in which u_r shows the velocity and direction of discharge of the water relative to the velocity and curve of the bucket, and is tangent to the curve of the bucket at the point of discharge *B*. u' shows the relative velocity of the periphery of the runner. The resultant of the two velocities is the absolute velocity with which the water is discharged from the turbine, and is shown in magnitude and direction by the arrow u''.

Now, at part gate the quantity of water discharged is less than that at full gate and hence u_r must also be less since the cross-section of the passage must be filled. u' remains unchanged and hence the resultant u_r will be increased with a corresponding waste of energy and loss in efficiency. This is an unavoidable loss in a wheel operating under part load, and makes it impossible to maintain high efficiencies of operation by any design of the regulating gates. This loss does not appear in the impulse wheel since the velocity with which the water leaves the bucket is, theoretically at least, not influenced by the quantity.

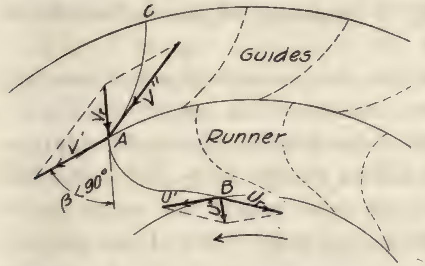


Fig. 193.—Graphical Representation of Low Speed Inward Flow Turbine (see page 287).

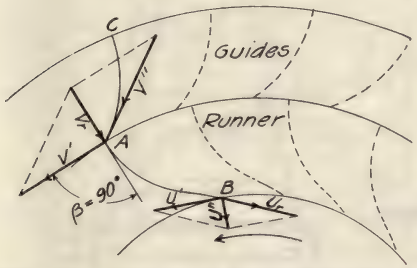


Fig. 194.—Graphical Representation of Medium Speed Inward Flow Turbine (see page 287).

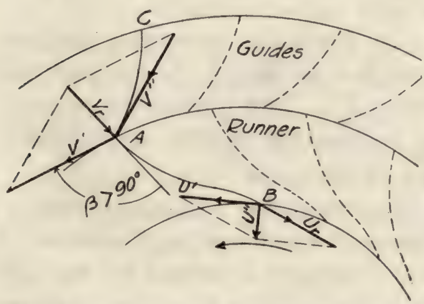


Fig 195.—Graphical Representation of High Speed Inward Flow Turbine (see page 287).

The third requirement is partially satisfied by gradually expanding the draft tube from the wheel to the point of discharge. This will recover only the component of the residual velocity in the axial direction. The larger component of the residual velocity however tends to produce a rotation of the water column in the draft tube, and is not recovered by any present design.

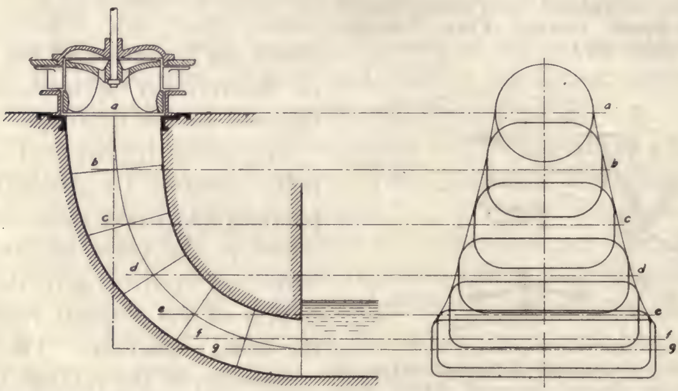
The fourth requirement is self-evident.

136. Graphical Relation of Energy and Velocity in Reaction Turbine.—The relations of the changes in velocity and in energy in the passage of water through a reaction turbine and its draft tube are graphically shown in Fig. 198, page 291.

Figure 196, page 290, shows the cross-section of a radial inward flow reaction turbine with a concrete draft tube. The cross-sections of

the draft tubes at various points are shown in Fig. 197 from which it will be seen that the draft tube of this turbine gradually changes form and increases in cross-section in order that the velocity of flow may be gradually decreased from the point of discharge of the turbine to the end of the draft tube.

The changes in absolute velocity in the passage of water into and through the turbine and draft tube are shown by line V, V_1, V_2, V_4, V_5; the height of the ordinates at these points shows the approximate absolute velocities at such points in the flow. The absolute velocity is a maximum at or near the point where the water enters the runner and is decreased as greatly as practicable at the point of



Figs. 196, 197.—Expanding Concrete Draft Tube for Reaction Wheel * (see page 289).

its discharge into the draft tube. By gradually increasing the area of the draft tube, an additional reduction in velocity is obtained, the water finally issuing with a velocity V_5. The maximum velocity, measured by the ordinate V_2, is, in reaction wheels, considerably below the spouting velocity ($\sqrt{2gh}$) of the water.

In its flow through the wheel, the velocity of the water relative to the bucket increases and becomes a maximum at the outlet of the wheel. This increase in relative velocity is shown by the line V_2, V_3.

The energy transformation which takes place during the change in velocity is illustrated by the dotted line marked "Energy transformation" which begins at a maximum of 100 per cent. at the en-

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* Turbinen and Turbinenanlagen, Viktor Gelpke, page 61.

trance of the wheel; is decreased by friction, leakage, shocks, etc., by about sixteen per cent. under full gate conditions. The energy is transformed into useful work in the wheel by the reactional pressure created by the passage of the water and about eighty per cent. of such energy is utilized, the remaining four per cent. being rejected in the discharge from the draft tube with a slight recovery of velocity energy as before described.

137. **Homologous Wheels.**—The diagramatic representations of relative speeds of wheels of different types (Figs. 193, 194 and 195, page 289) are general and each will apply to all wheels of similar design and of all practicable sizes. If therefore a series of wheels

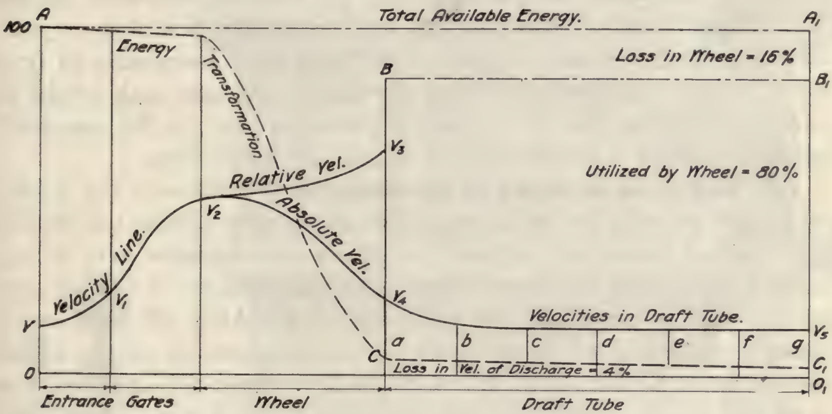

Fig. 198.—Graphical Relation of Velocity and Energy in the Flow Through a Reaction Turbine With Draft Tube (see page 289).

is constructed on the same general design with all parts in proportion, so far as practicable, they are termed wheels of "homologous design," and constitute a series of wheels which, if really homologous in both design and workmanship, will have common characteristics and hence common relations between their diameter D, and their discharge q, power P, speed n, and efficiency e, under any given effective head h. These important variables are somewhat independent but have certain interrelations which can be determined by experiment from any wheel of the series, and are expressed by coefficients from which their corresponding values for a wheel of the series of any given diameter can be determined for any given head and speed by application of principles which are hereafter discussed.

Quantitatively, the coefficients of a series of wheels will vary accordingly as the design of the series is for low, medium or high

speed, or for low, medium or high power; and in the selection of machinery for a water power project, in order that the best results may be attained, the turbine must be selected with due regard to the conditions under which it is to operate. In other words, the turbine, for the best results, can be advantageously used only under conditions which closely approximate those for which it was designed. The characteristics of wheels are important, therefore, as a basis for both their design and selection.

In general, the wheels of any manufacturer, designated as of a certain type, are usually practically homologous. Thus, all the sizes of the "Samson" turbines (see Fig. 199, page 293) of James Leffel & Company, and all sizes of the "Smith" turbines of the S. Morgan Smith Company, are approximately homologous turbines, the larger wheels of each type being built essentially in proportion to the smaller wheels of the same type, and each wheel of each type having the same characteristics as near as the manufacturer has found it practicable to design and build them.

138. Degree of Accuracy in Homologous Wheels.—In the manufacture of wheels, the inaccuracy of workmanship affects the results which can be secured, and no two wheels can therefore ever be exactly homologous in construction. The inaccuracies in casting, the inequalities in shrinkage, the difference in finishing, all combine to make variations more or less great in the comparative results which will actually be attained. Even two wheels from the same pattern will never give identical results in operation under the same conditions. The degree of similarity with which characteristic results will obtain will vary with the skill, workmanship and care entailed in their manufacture and in their installation. Usually the speed, power, discharge and efficiency of the best wheels are dependable within $\pm 2.5\%$. In a series of standard wheels, where great care is taken to secure homologous construction, the difference in characteristic results between different wheels of the same series may vary as much as $\pm 5\%$, and with careless workmanship, even greater variations will result. In series of standard wheels, homologous design is not always followed except in a general way, and occasionally some wheels of the series have been made more or less special and are introduced into the series so as to complete the list of sizes, but must be run under conditions of speed, power and discharge somewhat different from other wheels of the series. This condition is, however, not usual and most standard wheels do not vary more than $\pm 5\%$ from the average of the series as will be hereafter shown.

Fig. 199.—Turbines of Homologous Design Manufactured by James Leffel and Company (see page 292).

The departure of a series of standard wheels of homologous design from the average discharge is shown graphically by Fig. 206, page 310 and by Table 27, page 309, and the departure of the same series from the average power is shown graphically by Fig. 214, page 320, and by Table 31, page 319. The experiments from which these results were obtained were made over twenty years ago, and

present practice has improved materially. Some of the standard series yet in the market will however still show departures as great as shown by these tables, although in many cases the departure is not one-half so great.

139. **Speed Relations.**—In the tangential wheel the jet of water enters the bucket tangentially to the circumference on which the velocity of the wheel is measured, and with a velocity $v'' = c' \sqrt{2gh}$ slightly less than the spouting velocity v, of water under the head h. If the friction and windage of the wheel are negligible, the wheel may acquire a velocity v', measured on the circumference at which the jet is applied, and approximately equal to that of the jet v''. In practice, the maximum velocity v', of the bucket will always be consequently less than the velocity v'', of the jet, by the amount of velocity lost in overcoming the friction and windage of the wheel.

In the reaction wheel, the jets of water entering the wheel have a velocity of v'', considerably less than the spouting velocity v of water, and do not enter the buckets tangentially to the direction of motion of the buckets but at an angle depending upon the design of the wheel. The maximum velocity of the wheel may therefore be less or greater than the absolute velocity of the water which impels it, depending upon the design and friction losses.

In any water wheel with the ideal condition under which the velocity of the bucket and the component velocity of the water are the same, there will be no pressure of the water against the bucket, and the wheel under these conditions will receive no energy and can therefore deliver no power. As soon as resistance occurs, the speed of the wheel is checked, and under reduced speed, the kinetic energy of the jet or jets, according to the circumstances of design, is converted into pressure which transfers the energy of the water to the wheel by means of which the wheel is enabled to overcome both friction and resistance from the useful work which it is designed to accomplish. This impulse or pressure increases as the speed of the bucket decreases until the maximum pressure results when the bucket is at rest, in which case also no work is done. The speed of a turbine (see Fig. 207, page 311), therefore, may vary with the resistance from no speed or zero revolutions per minute to a maximum speed called the "runaway" speed (see Sec. 141) and dependent on the design of the wheel and on the head under which the machine works. This maximum speed is modified by friction and windage and will be further reduced if the wheel is connected

to other machinery which must also be rotated or moved against a friction load.

At some speed between these extremes, the maximum amount of power from a given turbine will be obtained (see Fig. 208, page 313). That is to say,—at a certain fixed speed, the maximum work or the maximum efficiency of a given wheel will be obtained, and at any speed below or above this speed, the power and efficiency of the wheel will be reduced. The speed for maximum power and for maximum efficiency, however, may or may not be the same. These conditions vary considerably according to the type and design of the wheel considered and also according to the gate opening at which the wheel may be operated. In calculating the velocity of tangential wheels, the diameter of the wheel is measured on the circumference at which the center of the jet or jets is applied (see Fig. 159, page 249). In the reaction wheel, such a circumference is indeterminate, and the diameter is measured at some point on the periphery of the impeller.

Many reaction wheels vary in diameter at various points on the periphery (see *B*, Fig. 158, page 247), and there is no uniform practice among manufacturers in designating such diameters so that the diameters used in the discussion of the wheels of various manufacturers and the functions based thereon are in accordance with the practice of each maker and are not strictly comparative unless similar. The principles discussed are however equally true if based on any actual diameter or any simple function of the same. The diameter chosen simply influences the magnitude of the derived coefficients and not their character or relations. The principle holds therefore in each case regardless of the method of measurement except for the purpose of comparison between wheels of various makers in which case similar diameters must be used.

The velocity of the periphery of the turbine may be considered as a function of the velocity due to head, in which case the relations will be expressed by the formula:

(84) $$v' = \phi \sqrt{2gh} \text{ from which}$$

(73) $$\phi = \frac{v'}{\sqrt{2gh}} = \frac{v'}{v}$$

These relations hold for all values of v' from its maximum speed without load to its minimum value of 0 when the wheel is stationary. In consequence, the relative value of ϕ will vary from 0 to a maximum; but there is only one speed for which the turbine is de-

signed to give the highest efficiency, and the values of v' and ϕ for such speed may be represented by a subscript thus

$$v_e' \text{ and } \phi_e$$

to indicate that these are the values at which maximum efficiency will obtain.

140. Relation of Speed to Diameter.—The velocity of the periphery of the runner of a turbine may also be expressed in terms of the diameter of the wheel D, and the number of revolutions per minute n.

$$v' = \frac{D\pi n}{12 \times 60} = \frac{3.1416\, Dn}{720} \tag{85}$$

Combining equations (73) and (85) it follows that:

$$\phi = \frac{3.1416\, Dn}{720 \times 8.025 \sqrt{h}} = .0005437 \frac{Dn}{\sqrt{h}} \tag{86}$$

and also that:

$$n = \frac{\phi \sqrt{h}}{.0005437 D} = \frac{1840\, \phi \sqrt{h}}{D} \tag{87}$$

As equation (86) is general, a speed coefficient may be deduced as follows:

$$\frac{Dn}{\sqrt{h}} = 1840\, \phi = \triangle \tag{88}$$

If the head is one foot ($h = 1$), equation (88) will reduce to:

$$Dn_1 = 1840\, \phi = \triangle \tag{89}$$

As equation (89) is general, it follows that for any fixed values of ϕ there is a corresponding fixed value of $\triangle$ and at these values the diameter of any wheel of a series multiplied by its corresponding revolutions per minute at any fixed head is constant.

If in equation (89) the diameter D is one inch $D = 1$ and the equation becomes:

(90) $n_1 = 1840\phi = \triangle =$ the revolutions per minute of a one inch wheel under one foot head.

When the value of ϕ in equation (89) is the speed ratio for the maximum efficiency of the wheel ϕ_e, then the corresponding value of $\triangle$ will be $\triangle_e$ or the value of $\triangle$ for maximum efficiency.

From the above considerations, it also follows that as in any homologous series of turbines, the wheels are designed to run at the same relative velocity; therefore when operating at the same speed ratio (ϕ constant)

$$\triangle = \frac{Dn}{\sqrt{h}} = \frac{D_d n_d}{\sqrt{h}} = \frac{Dn_h}{\sqrt{h_h}} = \frac{D_x n_x}{\sqrt{h_x}} \tag{91}$$

That is to say: *In any homologous series of turbines, operating at a fixed speed ratio, the product of the diameter of any wheel* D, *and its proper number of revolutions per minute* n, *divided by the square root of the effective head* $\sqrt{h}$, *under which it is to operate will equal the same speed constant* $\triangle$.

From equation (91), it follows that

$$n_x = \frac{Dn\sqrt{h_x}}{D_x\sqrt{h}} \tag{92}$$

From this equation the comparative speed or comparative number of revolutions n_x, for any wheel of diameter D_x, at any head h_x, can be obtained if the revolutions n, of any other wheel of the series at head h, and of diameter D, is known.

If in equation (92), $D = D_x$, the equation reduces to

$$n = \frac{n_x\sqrt{h}}{\sqrt{h_x}} \text{ or } \frac{n}{\sqrt{h}} = \frac{n_x}{\sqrt{h_x}} \tag{93}$$

That is to say, *the comparative speed of any wheel will be in direct proportion to the square root of the head under which it acts.*

If in equation (93), $h_x = 1$, the equation reduces to

$$n = n_1\sqrt{h} \tag{94}$$

From which it follows that *the revolutions* n, *of a wheel for any head* h, *are equal to the revolutions* n_1, *for one foot head multiplied by* $\sqrt{h}$.

EXAMPLES

(1) If a series of turbines are designed to operate with

$$\triangle_e = 1400$$

the corresponding value of ϕ will, from equation (88) be

$$\phi = \frac{\triangle}{1840} = \frac{1400}{1840} = .76$$

(2) The number of revolutions per minute of a 40-inch wheel of this series under one foot head will be, from equation (89)

$$n_1 = \frac{\triangle}{D} = \frac{1400}{40} = 35$$

(3) The number of revolutions per minute of the same wheel under 16 foot head will be, from equation (94)

$$n = n_1\sqrt{h} = 35\sqrt{16} = 140$$

(4) The speed of a 25-inch wheel of the same series under 16 foot head, will be, from equation (91)

$$\frac{Dn}{\sqrt{h}} = \frac{D_d\, n_d}{\sqrt{h}} \text{ then } \frac{40 \times 140}{\sqrt{16}} = \frac{25 n_d}{\sqrt{16}}$$

From which $n_d = 224$

141. Runaway Speed of Turbines.—In tangential wheels and other wheels in which the buckets are located at the periphery and are small in comparison to the wheel diameter (see Fig. 115, page 208, Diagrams C and D), the normal diameter is measured at a constant and easily determined position. In such wheels, the peripheral velocity at maximum speed will be less than the velocity of the jet acting on the wheel, which is essentially the spouting velocity of water (see Curve *A,* Fig. 200, page 299). That is

$$\phi_{max.} < \text{unity and } \triangle_{max.} < 1840$$

In reaction wheels, on account of the point of measurement of the diameter of the wheel, the angle of application of the water jets, and the fact that the circumference of resultants of the forces of the applied jets falls a considerable distance within the circumference of the wheel, such wheels commonly have a maximum peripheral velocity in excess of the spouting velocity of water due to the head under which they act (see Curve *B,* Fig. 200, page 299). That is

$$\phi_{max.} > \text{unity and } \triangle_{max.} > 1840$$

The runaway speed of wheels becomes of importance when considering the design of rotary machinery to be connected thereto. If such machinery is built only of sufficient strength to stand the regular speed for which the unit is designed to run, it may be greatly overstressed and perhaps destroyed if through any mischance the governor should be broken or disabled and the gates of the wheel should open and the turbine acquire its runaway speed. In several cases, electric generators have been destroyed by such accidents, and such generators should therefore be built for the highest speed under which they may be called upon to operate. Table 26, page 300, gives the results of various tests of tangential and reaction wheels in which are given the most efficient speed ratio of the wheel ϕ'_e, the runaway speed ratio ϕ_{max} and the ratio of $\frac{\phi_{max}}{\phi_e}$ which will show the percentage of over speed at which the wheel may operate under accidental conditions. The tangential wheels given in this table are small wheels tested in university laboratories. The re-

action wheels were mostly tested at the Holyoke testing flume. Tangential wheels are usually operated under such high head conditions that changes in head due to variations between high and low water will not materially alter their speed ratio. In reaction wheels operating under low heads, it is sometimes necessary to fix the speed ratio at less than ϕ_e for maximum heads, so that it will operate efficiently under average conditions. In such cases the ratio ϕ_{max} to the operating value of ϕ under the high head conditions con-

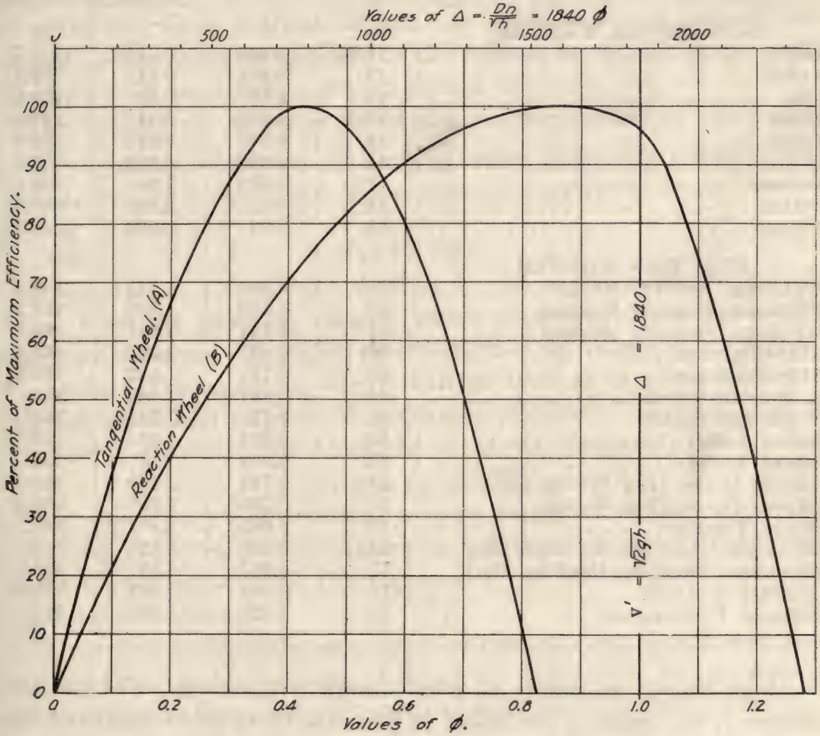

Fig. 200.—Relation Between Values of ϕ and Efficiency for a Tangential and for a Reaction Wheel (see page 298).

siderably exceed the limits given in the table. This table therefore cannot be used as a safe guide for design but the conditions must be considered for each individual case.

142. **The Efficient Speed of Turbines.**—In the tangential wheel, the position of the bucket in front of the jet is such that whatever the speed of the wheel may be, the water of the jet will enter the bucket

TABLE 26.

Relation of Speed Ratios Under Operating and Runaway Conditions.

Wheel.	Diameter inches	Value of φ Maximum Efficiency φ_e	Value of φ Runaway Speed φ_{max}	Ratio $\frac{\varphi_{max}}{\varphi_e}$ Per cent.
TANGENTIAL WHEELS				
DePuy	14.74	0.465	0.784	168.7
Leffel	12.	0.404	0.848	209.8
Hug	24.	0.46	0.78	169.6
Doble	14.6	0.397	0.747	188.4
Doble	12.	0.443	0.828	186.9
Pelton	12.	0.508	0.882	173.5
Pelton	12.	0.482	0.90	186.5
Pelton	18.	0.43	0.881	205.0
Pelton	18.	0.44	0.894	203.0
REACTION WHEELS				
Wellman, Seaver, Morgan	28.	0.800	1.215	151.9
Wellman, Seaver, Morgan	30.	0.782	1.238	158.1
Wellman, Seaver, Morgan	31.	0.781	1.285	164.5
Allis-Chalmers	30.	0.815	1.358	166.6
Allis-Chalmers	30.	0.75	1.332	177.5
S. Morgan Smith	30.	0.767	1.188	155.
S. Morgan Smith	48.	0.75	1.224	162.
James Leffel	50.	0.868	1.365	157.2
James Leffel	35.	0.899	1.40	156.
Dayton Globe Iron Works	44.	0.781	1.250	160.
Dayton Globe Iron Works	60.	0.759	1.263	166.4
Victor Turbine	48.	0.685	1.113	162.5
Hercules (Holyoke Machine Co.)	42.	0.665	1.120	168.
Hercules (Holyoke Machine Co.)	51.	0.677	1.151	170.
Original Francis	112.	0.652	0.988	151.5
Frances Fourneyron	81.	0.627	1.332	213.

"without shock" as nearly as good design will permit. For the best efficiency, the speed of the wheel in this case must be so regulated that the water will leave the buckets "without velocity" as nearly as practicable. In order to accomplish this as shown in Section 131, the velocity of the bucket v', measured at the point of application of the jet must be one-half the velocity of the jet:

$$v' = \frac{v}{2} \text{ or } \phi = \frac{v'}{v} = .5$$

Practically as friction intervenes and v' at its maximum can seldom have a velocity at runaway or no load speed of more than eighty-four per cent. to ninety-two per cent. of v, the actual value of v' for the

greatest efficiency of the tangential wheel will be from forty-two per cent. to forty-six per cent. of v; that is

$$\phi_e = \frac{v'_e}{v} = .42 \text{ to } .46$$

In the reaction turbine, from the requirement that *the water must enter the turbine without shock,* as discussed in Section 129, it is evident that the maximum efficiency of these turbines can be attained only when the relative velocities of the wheel and of the water are such as to admit the water without impact. In other words, a correctly designed turbine must be operated at the relative speed for which it was designed in order to secure the maximum efficiency.

The velocity v'', of water entering a reaction turbine is never equal to the spouting velocity $v = \sqrt{2gh}$, of water under the effective head but has some velocity which is less than, but always a simple function of v or

$$v'' = c''v \tag{95}$$

For this reason the relative velocity in the reaction turbine can be based upon the spouting velocity which is always determinate when the head is known. In order, therefore, that the velocity of the bucket v', at the entrance to the wheel shall be such as to admit the water without shock and to give the maximum efficiency, v_e' must bear the constant relation to v for which the wheel was designed; that is

$$v_e' = \phi_e v = \phi_e \sqrt{2gh}$$

The velocity v', of the turbine may be measured at any point which the designer elects, and as long as the relative velocity of turbine and water is such as to fulfill the requirement of the designer, maximum efficiency will obtain.

In reaction turbines the maximum or runaway speed will vary from 110 per cent. to 160 per cent. of v and the value of v' for greatest efficiency will be from sixty per cent. to ninety-five per cent.; that is

$$\phi_e = \frac{v'}{v} = .60 \text{ to } .90$$

143. Practical Demonstration of the Relations of Speed and Efficiency.—From the previous consideration of the first principles of turbine design, it follows that in any turbine running under different heads but otherwise under the same physical conditions as to gate opening, setting, draft tubes, etc., the efficiency will remain constant provided the ratio of the velocity of rotation to the theoretical spouting velocity of the water under the given head remains the same. That is to say, *the efficiency of a wheel will remain constant under*

various heads as long as the value of ϕ (or of $\triangle$) remains constant. This principle is demonstrated (1) by experiments made on a twelve inch Doble tangential wheel under heads from 49 to 165 feet, and (2) by experiments made on a twelve inch Morgan-Smith reaction wheel, under heads varying from about 7.10 feet to about 3.54 feet The results of each set of experiments have been platted in Figs. 201,* page 303, and 202,† page 304, from which it will be noted that all experimental points lie fairly close to the mean curve; that the variations therefrom are probably due to experimental errors (principally, it is believed, in the determination of the relative velocities); and that the reduction in head shows no uniform decrease in efficiency. The complete experiments show that this principle is true under all conditions of gate as well as for the full gate conditions, illustrated in Figs. 201 and 202.

For considerable changes in head, the principle is not strictly true. Under high heads, the losses in the wheel become such a small proportion of the power generated that an increase in efficiency may be expected. In such cases, however, the maximum efficiency at all heads, while not the same, will be obtained at the same value of ϕ. That is ϕ_e is always constant for the same wheel.

144. The Discharge of a Turbine With a Fixed Opening.—The nozzle of a tangential wheel with a fixed opening or the gates of reaction wheels with fixed openings, constitutes simply an orifice or series of orifices, or short tubes, to which the ordinary hydraulic formula for discharge applies, if the wheel is stationary, just as it would to any other types of orifice: that is, the velocity of flow

$$v = \sqrt{2gh} \tag{42}$$

The discharge at the fixed opening will be equal to the area of opening multiplied by the velocity and a certain coefficient of discharge, dependent on the character of the orifice, i. e. from equation (11)

$$q = av = ac\sqrt{2gh} \tag{96}$$

When the wheel is in motion, the discharge through the gates or nozzles which are stationary will remain the same as if the wheel were stationary, provided the buckets of the wheel clear the nozzle suf-

* "The Relations of Experimental Results to the Theory of the Tangential Wheel," by H. J. Hunt and S. M. Johnson. (See Bul. 337, University of Wisconsin.)

† "Test of a 12-inch McCormick Turbine," an unpublished thesis by O. W. Middleton and J. C. Whelan.

ficiently not to obstruct or interfere with the flow, as in the case of the tangential wheel. Where the buckets revolve in practical contact with the gates, as in the reaction turbine, they usually affect the flow, and

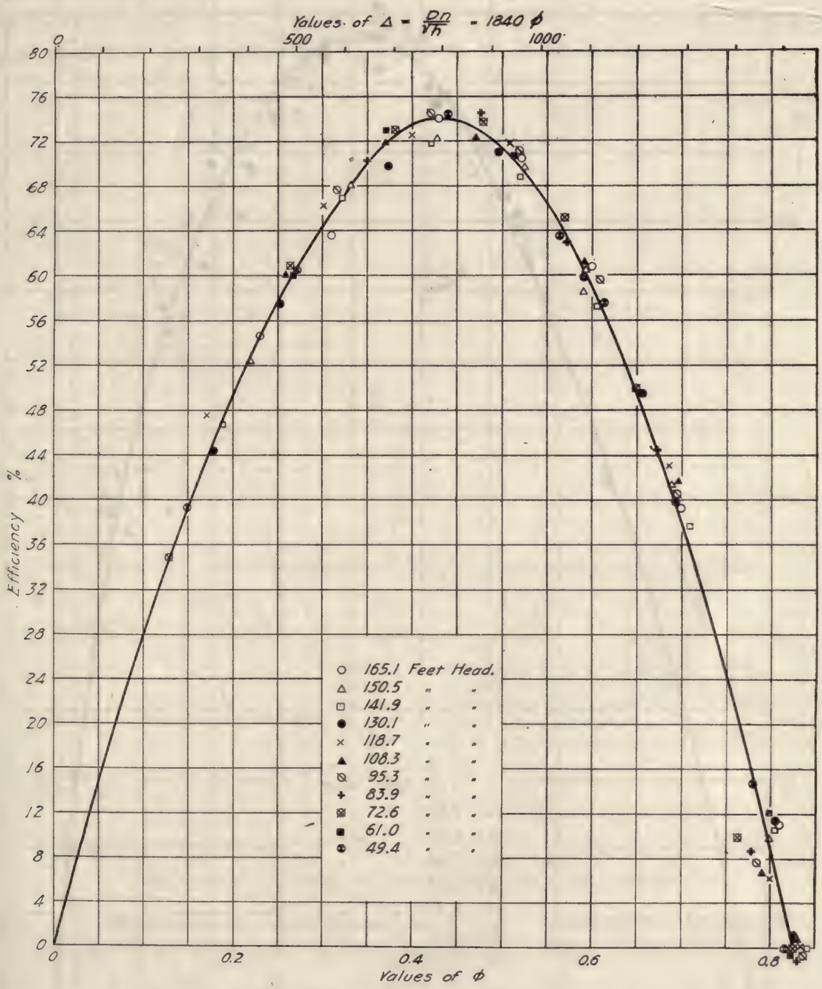

Fig. 201.—Relations Between Efficiency and Values of ϕ for a 12-inch Doble Tangential Wheel (see page 302).

the effect of their motion may be either to increase or to diminish the discharge and hence the value of the discharge coefficient as given in equation (96) is more or less modified. This condition is to be expected from the fact that every turbine is designed to admit water

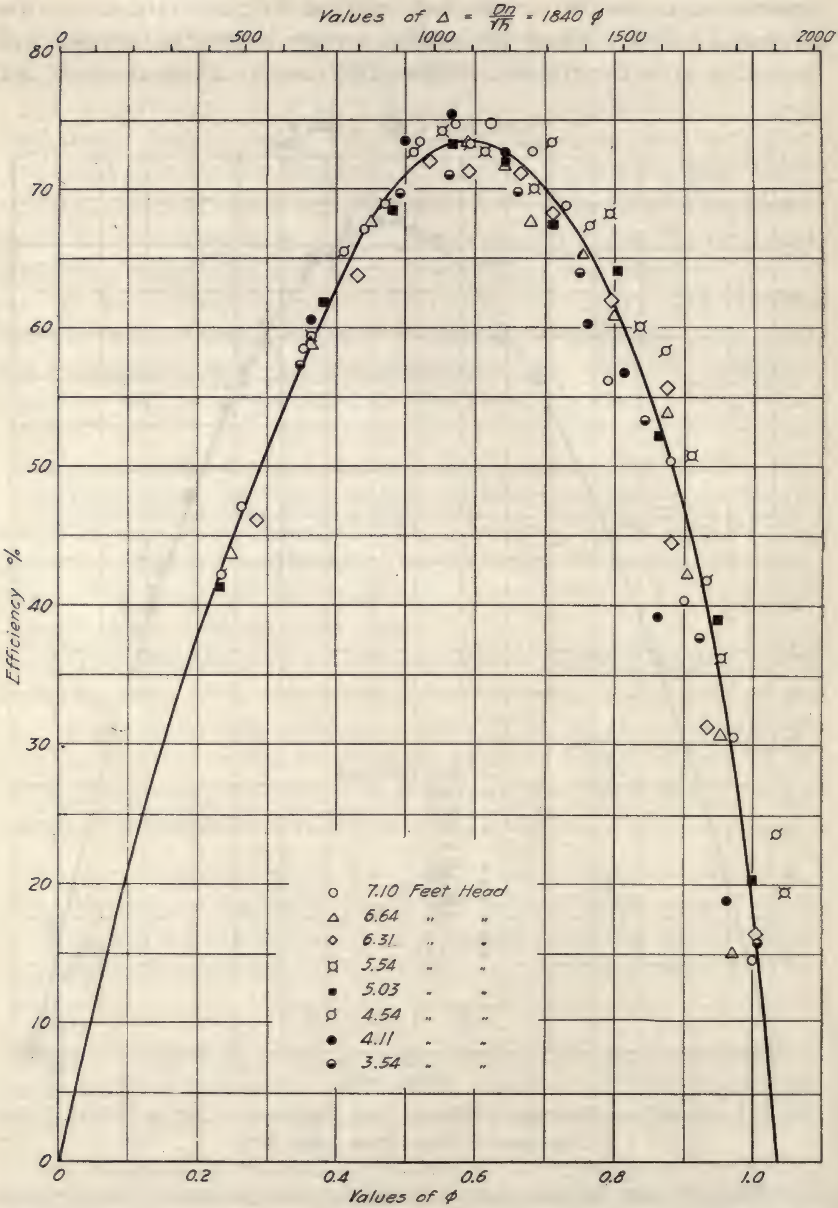

Fig. 202.—Relations Between Efficiency and Values of ϕ for a 12-inch Smith McCormick Reaction Wheel (see page 302).

without shock when the jet has a certain speed relation to the buckets, hence if the relative speed is changed either by increasing or decreasing the speed of the wheel the buckets will interfere with the discharge and shock will occur. In Fig. 203 are shown the relative discharges and speeds of various turbines as shown by Holyoke tests.

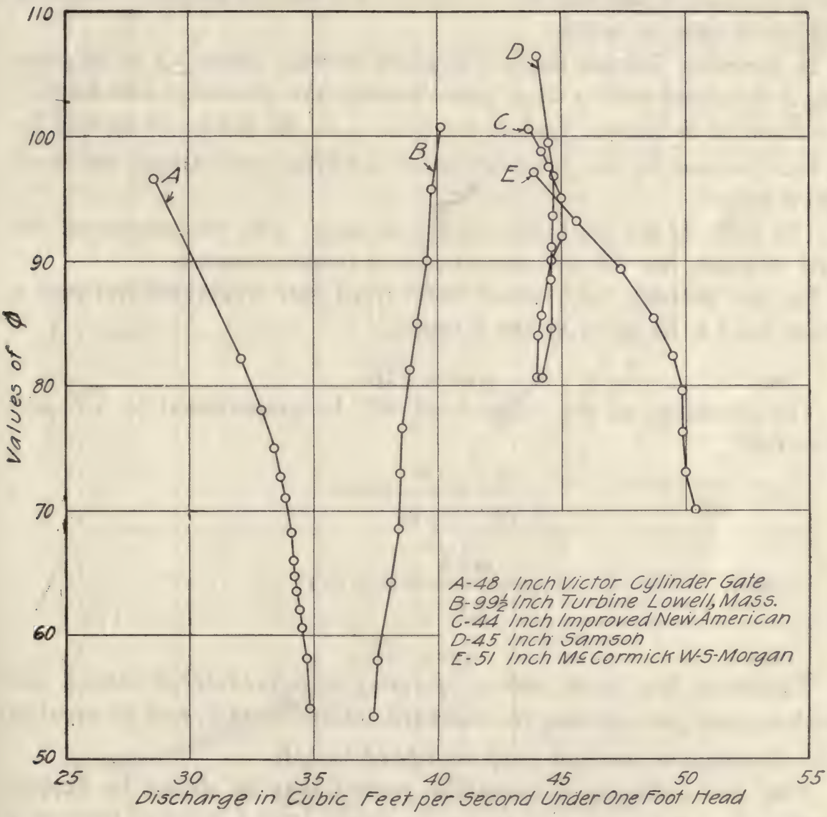

Fig. 203.—Relations Between Values of ϕ and Discharge for Various Turbines Under Full Gate Conditions. Platted From Holyoke Tests.

In some cases the discharge of a wheel increases as the speed increases (see discharge of Tremont turbine, Fig. 203). Sometimes the discharge decreases as the speed increases (see discharge of Victor and McCormick turbines, Fig. 203); and sometimes the discharge increases with the speed to a certain point and then decreases with a further increase in the speed (see discharge of Samson and New American wheels, Fig. 203).

In reaction turbines the discharge takes place first through the guide from which it passes into and through the buckets of the wheel. The relations of these two sets of orifices change as the speed of the wheel changes and while the value of c, in equation (96), varies with the speed, yet this value when once fixed for any given speed ratio ϕ, will remain constant and thus the conditions will remain similar to those of any short tube or orifice.

It therefore follows that, *in a given turbine operating at a given speed ratio and with a fixed gate opening, the discharge will be proportional to the square root of the head, i. e., the discharge divided by* $\sqrt{h}$ *is constant for the given turbine at the fixed gate opening and fixed speed ratio.*

The value of the discharge coefficient varies with the opening of the gate or gates, but for any one position it remain constant.

Let the discharge of a wheel under fixed gate conditions and with a given head h, be given by the formula

$$q = ca\sqrt{2gh} \tag{96}$$

The discharge at any other head will be proportional to $\sqrt{h}$ and therefore

$$\frac{q}{\sqrt{h}} = \frac{q_h}{\sqrt{h_h}} \text{ hence} \tag{97}$$

$$q = \frac{q_h\sqrt{h}}{\sqrt{h_h}} \text{ or if } h_h = 1 \tag{98}$$

$$q = q_1\sqrt{h} \tag{99}$$

Therefore, in a given turbine operating at a fixed speed ratio ϕ *and with a given gate opening the discharge at any head* h, *will be equal to the discharge at one foot head multiplied by* $\sqrt{h}$.

That this principle is essentially correct may be shown by experiment. Fig. 204, page 307, shows the results from a series of tests on a small McCormick turbine at full gate. Three sets of experiments are platted with values of ϕ equal to .35, .65 and .90 and for heads from about 4.25 feet to 7.1 feet. Fig. 205, page 308, shows the discharge of this turbine at various gate openings and under seven different heads. For the purpose of this diagram the discharges under each head have been reduced to the theoretical discharge at one foot head by equation (99). It will be noted from both Fig. 204 and Fig. 205 that all experiments where ϕ is the same lie close to the average line, and that

the departures from this line are simply those due to experimental errors. The results are sufficiently close, however, to show that the discharge under practical conditions essentially follows the law above expressed. Common practice under much greater ranges of head has demonstrated the truth of this principle.

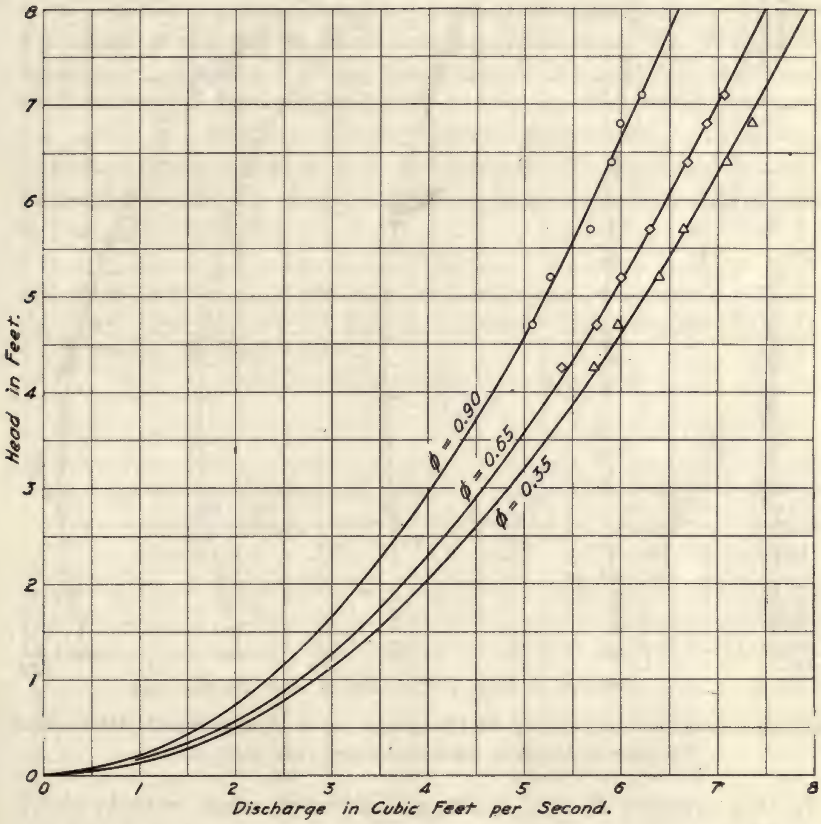

Fig. 204.—Relation of Head to Discharge for a 12-inch Smith McCormick Turbine (see page 306).

145. The Relation of Discharge to the Diameter of a Turbine.—In any homologous system of turbines, the diameters, heights and corresponding openings and passages being proportional, it follows that similar discharge areas under different gate conditions are proportional to each other and to the squares of any lineal dimensions. In such wheels, therefore, the area a, of the gate openings is proportional to

the square of the diameter of the wheel, and the equation may therefore be written:

(100) $$ca\sqrt{2g} = KD^2$$

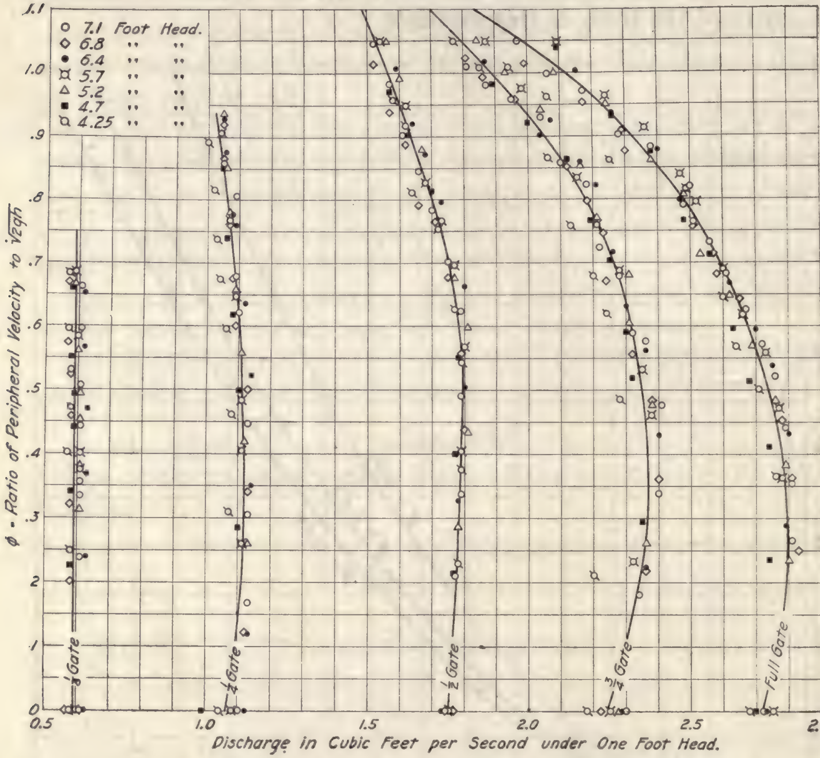

Fig. 205.—Relation of Velocity to Discharge for a 12-inch Smith McCormick Turbine at Various Gate Openings (see page 306).

In this equation K is a discharge coefficient which may be determined by experiment. Combining equations (96) and (100), it follows that:

(101) $$q = KD^2\sqrt{h}$$

If in equation (101), the size of the turbine be reduced to one inch ($D = 1$) and if the head be made one foot ($h = 1$), the equation becomes

(102) $$q = K$$

That is, *the discharge coefficient* K, *is the discharge of hypothetical one inch wheel under one foot head.*

From equation (101), by transposition there results

$$D = \sqrt{\frac{q}{Kh^{1/2}}} \qquad (103)$$

EXAMPLE

If the discharge coefficient for a series of wheels be $K = .006$, the discharge of a 40-inch wheel under one foot head will be, from equation (101)

$$q = KD^2 = .006 \times 1600 = 9.6 \text{ cubic feet per second}$$

The discharge of the same wheel under any other head, will from the same equation be proportional to the square root of the head $\sqrt{h}$, therefore under a sixteen foot head a 40-inch wheel of this type will have a discharge

$$q = 9.6 \times \sqrt{16} = 38.4 \text{ cubic feet per second}$$

That equation (101) is both theoretically and practically correct is shown by the data in Table 27, which is also graphically represented in Fig. 206,* page 310.

TABLE 27.

Discharge of Thirteen Water Wheels of the Same Manufacture but of Different Diameters, as Determined by Actual Tests, Compared With Value Computed by the Formula:

$$q = KD^2 \sqrt{h} \text{ in which } h = 13, K = .0172$$

DISCHARGE.

No.	Diameter in inches.	Reduced from actual tests, Cu. ft. per Sec.	Computed (Mean Curve) Cu. ft. per Sec.	Variation from Computed Discharge Cu. ft. per Sec.	Per cent. Variation from Computed Discharge.
1	9	5.17	5.02	+0.15	+2.99
2	12	8.79	8.92	—0.13	—1.46
3	15	13.85	13.93	—0.08	—0.57
4	18	18.85	20.07	—1.22	—6.08
5	12	29.07	27.32	+1.75	+6.41
6	24	35.31	35.68	—0.37	—1.04
7	27	47.81	45.16	+2.65	+5.87
8	30	54.15	55.75	—1.60	—2.87
9	36	77.33	80.28	—2.95	—3.67
10	39	93.51	94.22	—0.71	—0.75
11	42	107.73	109.27	—1.54	—1.41
12	45	128.53	125.44	+3.09	+3.10
13	51	161.07	161.12	—0.05	—0.03

In this table are given the discharges determined from actual test of thirteen water wheels of various diameters. These results have been reduced to the common basis of the discharge at thirteen foot head.

* See paper "Notes on Water Power Equipment" by A. W. Hunking. Jour. Asso. Eng. Soc., Vol. 13, No. 4. April 1894.

The computed discharges at thirteen foot head on the basis of equation (101) are also given, as well as the percentage of variations of the actual from the theoretical discharges. The wheels were of the same make with inward and downward discharge. The departures or variations from the mean values, as determined by calculation, are due both to imperfections in the construction of the wheel and to errors in the tests. They are, however, seen to practically conform to the theoretical deductions. From this table and Fig. 206, are also shown the departures in discharge q, of a series of homologous turbines from the mean values for the series.

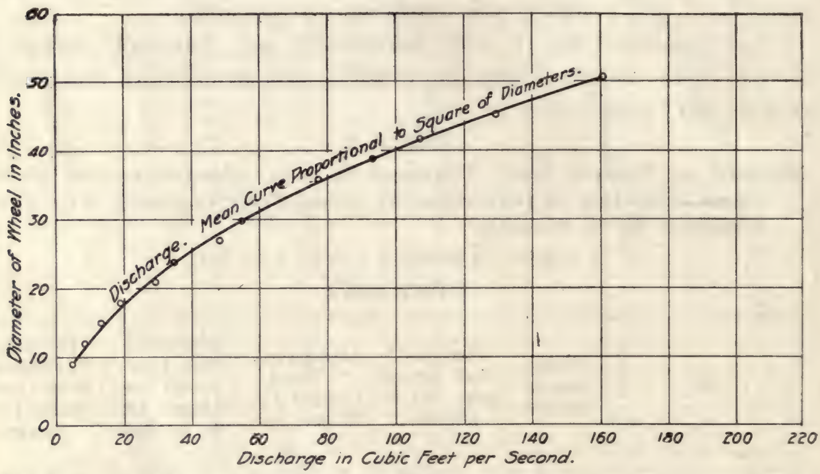

Fig. 206.—Relations of Discharge to Diameter in Reaction Turbine of Homologous Design (see page 309).

146. Power of a Turbine.—Work done by a turbine, measured by a friction brake, may be expressed by the product of resistance overcome through a given space. That is:

$$\text{Work} = \text{Resistance} \times \text{Space} \tag{104}$$

and power, which is simply a rate of performing work, may be expressed in terms of the factors of a friction brake as follows:

$$P = \frac{2\pi lwn}{33000} \tag{105}$$

In this equation, the factor $\frac{2\pi lw}{33000}$ may be regarded as a resistance factor, measuring resistance in foot pounds per minute overcome at each revolution; and the factor n, the number of revolutions per minute, may be regarded as a measure of space passed through. The product of these two factors equals the power.

The relation of speed to resistance in a forty-eight inch turbine at full gate under thirteen foot head, in terms of resistance and speed, is shown in Fig. 207 and Table 28. On the right hand of Fig. 207

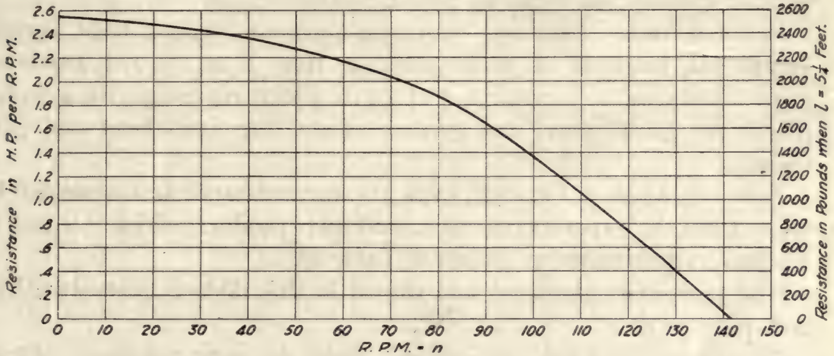

Fig. 207.—Relation of Speed to Resistance in 48-inch Turbine Under 13-foot Head (see page 312).

TABLE 28.

Actual Power of a Water Wheel.

Speed and Resistance of 48" Turbine at Full Gate and Under 13' Head.

R. P. M.	Resistance in Horse Power per Revolution per Minute.	Corresponding Actual Power of Turbine.
0	2.55	0
10	2.52	25.2
20	2.48	49.6
40	2.35	94.0
60	2.17	130.2
80	1.87	149.6
90	1.64	147.6
100	1.37	137.0
120	.74	88.8
130	.41	53.3
140	.06	8.4
141.6....................	0.00	0.0

is shown the actual brake load w, with a length l, of the brake arm of 5.25 feet. On the left hand, the corresponding friction factor $\frac{2\pi lw}{33000}$ is given.

With no resistance (except friction and windage), the peripheral speed or velocity v' of this wheel, on the periphery of its nominal diameter, is $v' = 1.1127\sqrt{2gh}$. Under this condition there is speed but

no useful resistance, and therefore no power. As resistance is applied, the velocity of the wheel is decreased, until finally the resistance is so great that the wheel is stopped. Here resistance is exerted but without motion, and hence no power results.

Between these extremes, both resistance and speed have certain values, the products of which increase from 0 at the two extremes above mentioned to a maximum of 149.6, which represents the greatest power the given wheel can generate under the fixed head and gate opening.

From Table 28 or Fig. 207, page 311, the resistance and speed can be taken from the experimental data, and their product, which is equal to the power, determined as shown in Table 28.

The power as calculated and shown in this table is platted in Fig. 208, page 313.

The discharge of the same turbine, under the same conditions of gate and head and at various speeds, and the corresponding theoretical power of the water discharged, is shown in Table 29.

TABLE 29.

Theoretical Power of Water Used by Wheel.

Speed and Discharge of 48" Wheel at Full Gate under 13' Foot Head.

R. P. M. n	Discharge in Cubic Feet per Second.	Theoretical Power of Water.
0	129.2	190.7
10	129.0	190.4
20	128.7	190.0
40	127.8	188.7
60	126.5	186.7
80	124.4	184.0
90	122.0	180.0
100	118.5	175.0
120	108.0	159.4
130	101.5	149.8
140	94.8	139.8
141.6	93.7	138.2

The power curve of the wheel shown in Fig. 208 is reproduced in Fig. 209, page 314, together with the curve of the theoretical power of the water which was discharged during the tests as calculated in Table 29. As the theoretical power of the water is in direct proportion to the discharge, this same power curve with a different scale (see scale to the right) will show the actual discharge of the turbine at full gate and at different rates of speed. If the power of the wheel at

different speeds be divided by the theoretical power of the water discharged at the same speed (see Fig. 210, page 314, and Table 30), the results will give the efficiency of the wheel under the various conditions of speed. These are shown in detail in Table 30, and calculated efficiencies from this table are platted in Fig. 210.

Figures 209 and 210 are typical of all turbines under all gate openings and show that a turbine of certain design and given diameter, to

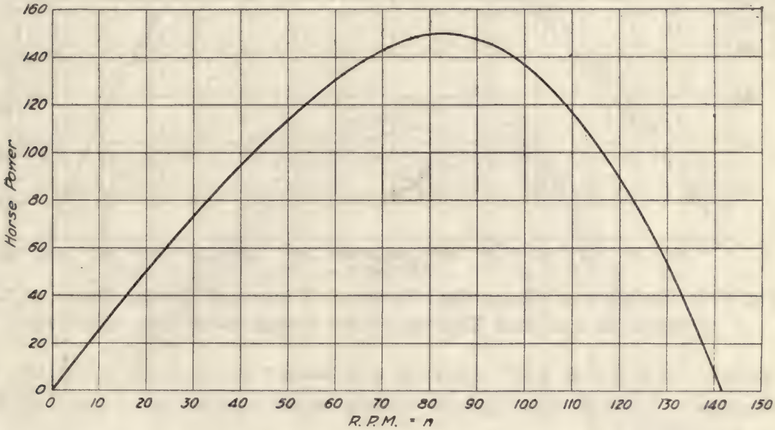

Fig. 208.—Relation of Power to Speed in a 48-inch Turbine Under 13-foot Head (see page 312).

TABLE 30.

Efficiency of Turbine.

Speed and Efficiency Relation of 48" Turbine at Full Gate under 13' Head.

R. P. M.	Actual H. P. of Wheel.	Theoretical Power of Water.	Efficiency of Wheel.
0	0	190.7	0
10	25.2	190.4	13.2
20	49.6	190.0	26.1
40	94.0	188.7	49.8
60	130.2	186.7	69.8
80	149.6	184.0	81.2
90	147.6	180.0	81.9
100	137.0	175.0	78.3
120	88.8	159.4	55.7
130	53.3	149.8	35.5
140	8.4	139.8	6.0
141.6	0.0	138.2	0.0

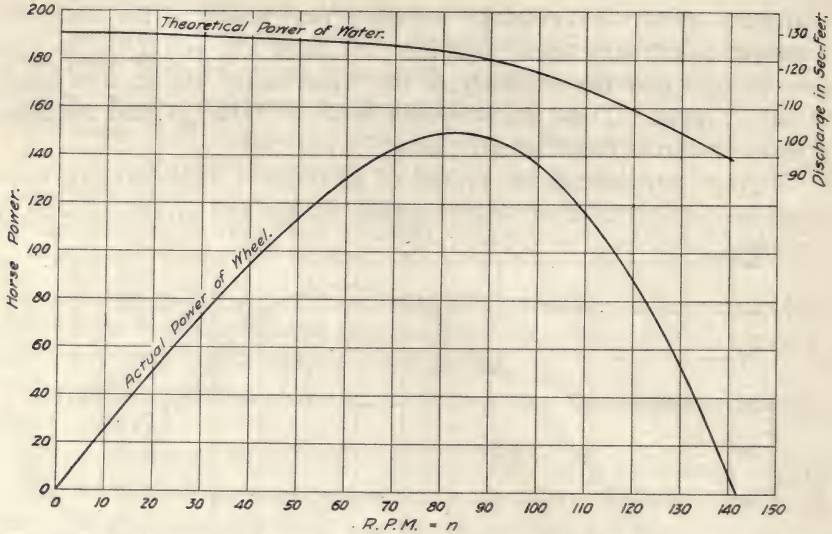

Fig. 209.—Relation of Theoretical Power of Water and Actual Power of Wheel to Speed in a 48-inch Turbine Under 13-foot Read (see page 312).

operate at a given gate opening and under given head with the best efficiency, or with the highest power, must operate at a fixed number of revolutions.

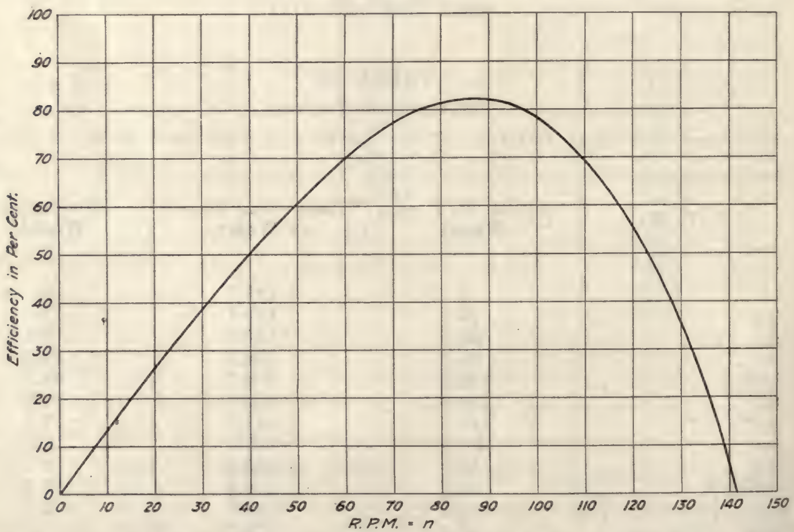

Fig. 210.—Relation of Efficiency to Speed in a 48-inch Turbine Under 13-foot Head (see page 313).

The speed or R. P. M. for highest efficiency and for highest power is not ordinarily the same in reaction turbines on account of the fact that the discharge varies somewhat with the speed, as shown by the Theoretical Power-Speed and Discharge-Speed Curve in Fig. 209, page 314. The speed for maximum power under thirteen foot head is about eighty-three R. P. M. for this wheel (see Fig. 209), while for maximum efficiency it is about eighty-eight R. P. M. (see Fig. 210, page 314).

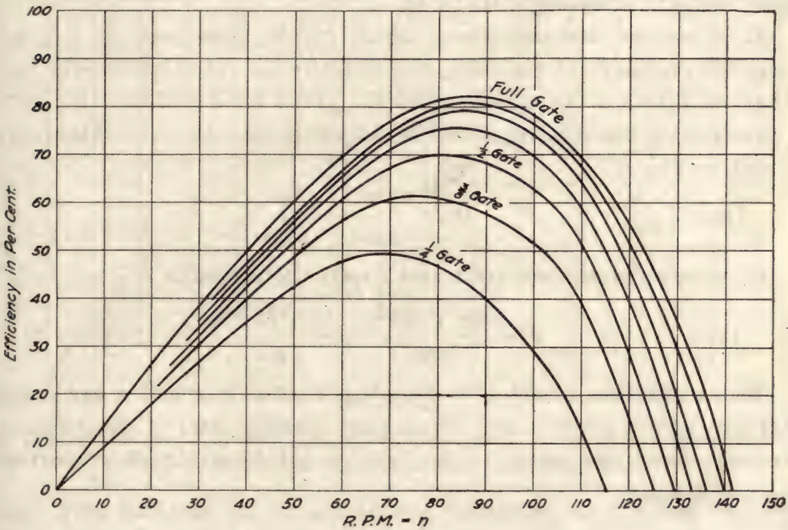

Fig. 211.—Relation of Efficiency and Speed in a 48-inch Turbine Under 13-foot Head at Different Gate Openings.

Figure 211, shows the efficiency-speed curve of the same wheel at different gate openings from which it will be noted that the highest efficiency at different gate openings does not occur at the same speed. If, therefore, this wheel was to be operated under certain gates for a major portion of the time it should be speeded in accordance with the conditions of operation. For example, if it is to be operated at full gate for most of the time, its speed should be eighty-eight R. P. M. for the best efficiency; but if it is to be operated at five-eighths gate most of the time, its speed should be eighty-three R. P. M. The gain in efficiency, due to this change in speed, in this particular case, would not be large. It should be noted that the R. P. M. should usually be determined by the speed at maximum efficiency for the particular capacity at which the wheel is to be operated rather than the speed for the maximum efficiency of the wheel.

Figure 212, page 317, shows the power-speed curves of the same wheel at different gate openings. This diagram shows that the maximum power for each gate opening does not occur at the same speed. The highest power for full gate conditions as shown, is obtained at eighty-three R. P. M. while the maximum power for less gate openings occurs at lower speeds that for five-eighths being at eighty R. P. M., for one-half gate being at seventy-six R. P. M. and for one-fourth gate being at sixty-eight R. P. M.

It is evident that the power which can be generated by any wheel depends on the head available, the quantity of water which will be discharged through the wheel under the given head and the efficiency of operation at the relative speed at which it may be run. Hence from equation (2)

$$\text{(106)} \qquad P = \frac{qwhe}{550} = \frac{qhe}{8.8}$$

Combining equations (96) and (106) there results

$$\text{(107)} \qquad P = \frac{caw\sqrt{2g}\,h^{\frac{3}{2}}e}{550} = \frac{ca\sqrt{2g}\,h^{\frac{3}{2}}e}{8.8}$$

From equation (107) it is apparent that if *c*, *e* and *a* are constant for any given turbine and fixed gate opening, and if the value of ϕ remains fixed, the power of the turbine will be in direct proportion to $h^{\frac{3}{2}}$ consequently

$$\text{(108)} \qquad \frac{P}{h^{\frac{3}{2}}} = \frac{P_h}{h_h^{\frac{3}{2}}}$$

Equation (108) may be transposed to

$$\text{(109)} \qquad P = \frac{P_h\, h^{\frac{3}{2}}}{h_h^{\frac{3}{2}}}$$

From which can be determined the power of the wheel at any given head and at the same relative speed, provided its power at any other head is known.

In equation (109) if $h_h = 1$, there results

$$\text{(110)} \qquad P = P_1 h^{\frac{3}{2}}$$

From which it follows that *in any turbine with a fixed gate opening, the power that can be developed at any head and the same relative speed will be equal to the power at one foot head multiplied by* $h^{\frac{3}{2}}$.

This principle may also be demonstrated experimentally. In Fig. 213, page 318, is shown the theoretical curve representing the relation between head and horse power of the twelve inch McCormick turbine

before mentioned. The turbine on which these experiments were made was small and the heads were limited, so that there is some variation from the theoretical curves, but the principle expressed by the general law is clearly shown.

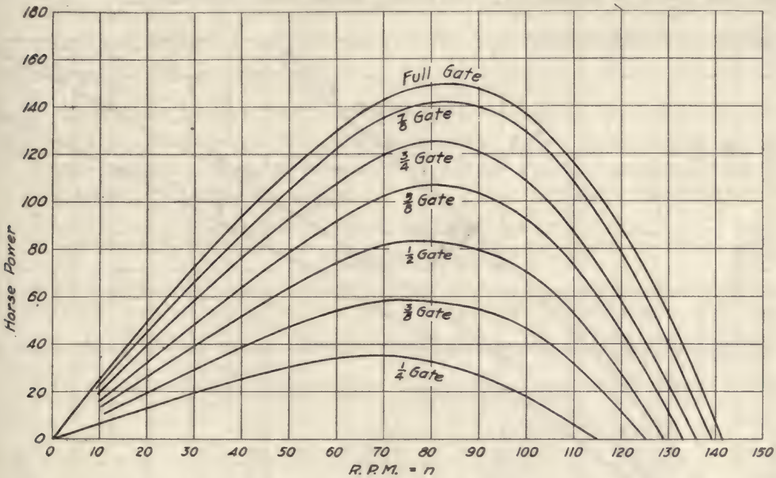

Fig. 212.—Relation of Power and Speed in a 48-inch Turbine Under 13-foot Head at Different Gate Openings (see page 316).

147. The Relation of Power to the Diameter of a Turbine.—By substituting the value of q, from equation (101) in equation

(106) $$P = \frac{qhe}{8.8}$$

it follows that

(111) $$P = \frac{D^2 h^{\frac{3}{2}} Ke}{8.8} = \left(\frac{Ke}{8.8}\right) D^2 h^{\frac{3}{2}}$$

As $\left(\frac{Ke}{8.8}\right)$ is constant for a given wheel, as long as ϕ is constant, this expression may be represented by a constant p which is the power coefficient and may be derived independently for each make of wheel or may be determined from the equation

(112) $$p = \frac{Ke}{8.8}$$

With this substitution (111) becomes

(113) $$P = pD^2 h^{\frac{3}{2}}$$

That is to say, *with wheels of homologous design, the power of any wheel, under the given head and the same relative speed, is in direct proportion to the square of its diameter.*

If in equation (113) a diameter of the wheel of one inch $D = 1$, and a head of one foot $h = 1$, be considered, there results

(114) $$P = p$$

From which it follows that *the power coefficient* p, *is the power of a wheel one inch in diameter, operating under one foot head.*

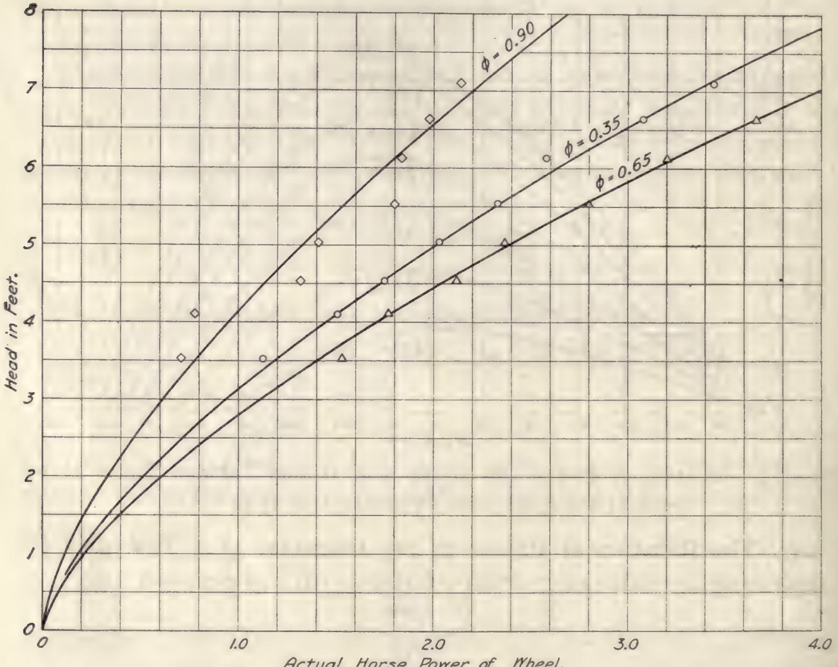

Fig. 213.—Relation Between Head and Horse Power in a 12-inch Smith McCormick Turbine (see page 316).

EXAMPLE

If the power coefficient for a series of turbines be $p = .004$, the power of a 40-inch wheel under one foot head will be, from equation (113)

$$P = pD^2h^{\frac{3}{2}} = .004 \times 1600 \times 1 = 6.4 \text{ horse power}$$

and the power of the same wheel at any other head will be in proportion to the three-halves power of the head $h^{\frac{3}{2}}$; therefore under a sixteen foot head, a 40-inch wheel of this type will produce the power as given by the expression

$$P = 6.4 \times 64 = 409.6 \text{ horse power}$$

That equation (113) is both theoretically and practically correct is demonstrated by the experiments tabulated in Table 31 and platted in Fig. 214, page 320. These data were taken from the paper by Mr. Hunking to which reference has previously been made. This table

and figure illustrate the relation between the theoretical power, as determined by equation (113), and the actual horse power of thirteen wheels of the same manufacture but different diameters, as determined by actual tests.

Table 31 and Fig. 214 still further illustrate the departure of a series of standard homologous turbines from the mean values given by the coefficients of the series.

TABLE 31.

Horse Power of Thirteen Water Wheels of the Same Manufacture but of Different Diameters, as Determined by Actual Tests, Compared With Values Determined by the Formula:

$$P = p D^2 h^{\frac{3}{2}} p = .00158 \quad h = 13$$

HORSE POWER.

No.	Diameter in inches.	From Tests.	Computed.	Variation from Computed H. P. in H. P.	Variation from Computed H. P. Per cent.
1	9	6.10	6.00	+0.10	+1.67
2	12	10.41	10.67	—0.26	—2.44
3	15	16.49	16.67	—0.18	—1.08
4	18	22.89	24.00	—1.11	—4.62
5	21	33.71	32.67	+1.04	+3.18
6	24	41.53	42.67	—1.14	—2.67
7	27	56.67	54.07	+2.60	+4.81
8	30	63.69	66.68	—2.99	—4.48
9	36	97.45	96.00	+1.45	+1.50
10	39	109.98	112.68	—2.70	—2.40
11	42	133.09	130.69	+2.40	+1.84
12	45	153.82	150.02	+3.80	+2.53
13	51	196.28	192.69	+3.59	+1.86

148. Relation of Turbine Speed and Power.—There are certain relations of speed and power which are important in the preliminary selection of a turbine for given work.

From equation (88), page 296, it follows that:

$$D = \frac{\triangle \sqrt{h}}{n}$$

From equation (113), page 317, it also follows that

$$D^2 = \frac{P}{Ph^{\frac{3}{2}}}$$

By equating the values of D^2, it follows that

$$(115) \qquad \frac{\triangle^2 h}{n^2} = \frac{P}{ph^{\frac{3}{2}}}$$

and therefore

(116) $$\triangle^2 p = \frac{n^2 P}{h^{\frac{5}{2}}}$$

The combined coefficient $\triangle^2 p$ is represented by the symbol $\mathfrak{P}$ which may be termed the specific power of the wheel.

(117) $$\mathfrak{P} = \triangle^2 p = \frac{n^2 P}{h^{\frac{5}{2}}}$$

When the head is equal to one foot, equation (117) becomes

(118) $$\mathfrak{P} = n^2 P$$

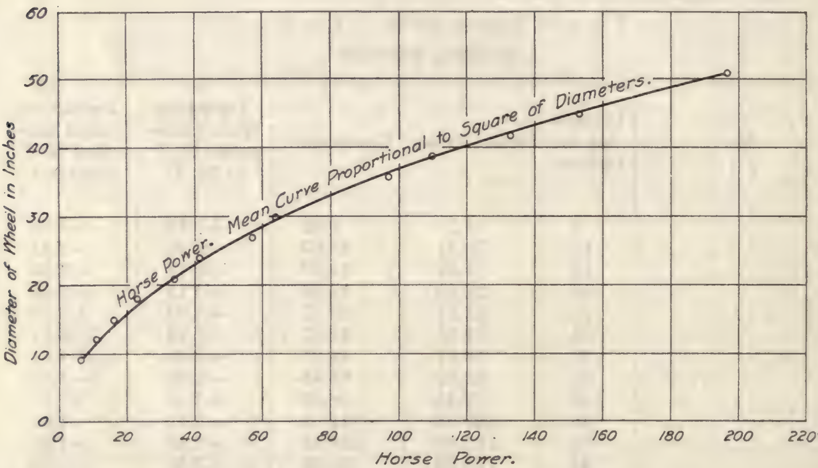

Fig. 214.—Relation of Power to Diameter in Reaction Turbines of Homologous Design.

and when the number of revolutions per minute also equal one, it follows that

(119) $$\mathfrak{P} = P$$

Which means that $\mathfrak{P}$ *is the power of a wheel of such diameter that it will make one revolution per minute under one foot head.*

American water wheel designers use a similar coefficient which is the square root of $\mathfrak{P}$ and is usually termed the "unity speed" and is represented by the symbol N_u.

(120) $$N_u = \sqrt{\mathfrak{P}} = n\sqrt{\frac{P}{h^{\frac{5}{2}}}}$$

If in equation (120) the power is taken at unity $P = 1$, and the head is taken at one foot $h = 1$, there results

(121) $$N_u = n$$

or N_u *equals the revolutions per minute of a wheel of such size that it will produce one horse power under a head of one foot.*

European turbine designers use the metric coefficient, which is usually termed the "specific speed" and is designated by the symbol N_s.

$$N_s = 4.46\ N_u = 4.46\ n\sqrt{\frac{P}{h^{\frac{5}{2}}}} \tag{122}$$

expressed in English units, or

$$N_s = n\sqrt{\frac{P}{h^{\frac{5}{2}}}} \tag{123}$$

if P and h are exressed in metric units. If in this equation the power and head are taken as unity

$$N_s = n \tag{124}$$

or *the specific speed of a wheel is defined as the number of revolutions per minute of a wheel of such size that it will produce one metric horse power under a head of one meter.*

The terms "unity speed" and "specific speed" are sometimes used interchangeably in the United States for the expression N_u which is also sometimes called the "type characteristic."

For the use of the water power engineer in the selection of turbines, the form $\mathfrak{P}$ is better adapted than the form N_u or N_s on account of the fact that $\mathfrak{P}$ varies directly with the power P of turbines, while N_u and N_s vary with $\sqrt{P}$.

If therefore the value of $\mathfrak{P}$ for the combination of power and speed relations is such that it is above the value of any available wheel, the conditions can be satisfied by the use of two or more wheels having such a common specific speed that their combined value will equal the required specific speed. For example, if the required $\mathfrak{P} = 20{,}000$, it is found that there is no wheel which has such a high specific speed. The condition can, however, be satisfied by

2 wheels with a specific speed $= 10{,}000 = 20{,}000$
3 wheels with a specific speed $= 7{,}333 = 20{,}000$
4 wheels with a specific speed $= 5{,}000 = 20{,}000$

149. **Practical Signification of Specific Power.**—From equation (119) specific power is defined to be the power of a wheel of such size that it will make one revolution per minute under one foot head.

$$\mathfrak{P} = \triangle^2 p = \frac{n^2 P}{h^{\frac{5}{8}}} \tag{117}$$

From equation (86)

$$n = \frac{720\,\phi\,\sqrt{2gh}}{\pi D} \text{ from which}$$

(124) $$n^2 = \frac{518{,}400\phi^2 2gh}{\pi^2 D^2}$$

and from previous equations (106) and (96)

$$P = \frac{qhe}{8.8}$$

$$\text{and } q = av'' = ca\,\sqrt{2gh}$$

Combining these equations, there results

(125) $$P = \frac{caeh^{\frac{3}{2}}\sqrt{2g}}{8.8}$$

Then combining (124) and (125)

(126) $$\mathfrak{P} = \frac{518{,}400\phi^2 2gh}{\pi^2 D^2} \times \frac{caeh^{\frac{3}{2}}\sqrt{2g}}{8.8\ h^{\frac{5}{2}}}$$

$$= \left(\frac{518{,}400\ \varphi^2\ 2g^{\frac{3}{2}}ce}{8.8\ \pi^2}\right)\frac{a}{D^2}$$

As all the terms of the expression within the parentheses are either constant or fixed by the design of the turbine, the expression becomes constant for a given design and therefore

127) $$\mathfrak{P} = c\frac{a}{D^2}$$

For impulse wheels, the area of discharge may be expressed in terms of the diameter *d*, of the jet as follows:

(128) $$a = \frac{\pi d^2}{4}$$

Equation (127) therefore becomes

(129) $$\mathfrak{P} = \frac{c\pi}{4}\,\frac{d^2}{D^2}$$

From which it is seen that *the specific power of an impulse wheel is proportional to the ratio between the square of the diameter of the jet and the square of the diameter of the wheel.*

For reaction wheels, the area of discharge may be expressed by the product of a function of the diameter *D* multiplied by the height of the speed gate *E*. That is:

(130) $$a = c'''DE$$

Hence equation (127) becomes

$$\mathfrak{P} = cc''' \frac{E}{D} \tag{131}$$

From which it is seen that *the specific power of reaction wheels of the same speed characteristics is proportional to the ratio between the height of the speed gate and the diameter of the wheels.*

150. Turbine Coefficients—Their Relations and Variations.—In the preceding discussion, various turbine relations have been expressed by means of certain coefficients which are common in value to all turbines of the same type or of the same homologous design and construction, but differ in value between turbines of different types. For the intelligent design or selection of turbines a definite conception must be had of the characteristics as representing turbines of different types, of what these coefficients indicate, their relation one to another and how they vary in turbines of the same and of different types.

In this connection, it seems advisable to again review the meaning and relations of the coefficients.

(42) $v = \sqrt{2gh} =$ *The spouting velocity of water*, or the total velocity v, which would be acquired by a jet of water under the head h.

(95) $v'' = c'' \sqrt{2gh} =$ *The actual velocity of water at entrance to the wheel*, where c'' depends upon the design and size of water passages through the wheel.

(85) $v' = \dfrac{\pi Dn}{12 \times 60} =$ *The peripheral velocity of the runner.*

(73) $\phi = \dfrac{v'}{v} =$ *The ratio of peripheral velocity of the runner to the theoretical velocity of the water, due to the head.*

ϕ therefore (through v') depends upon the angle of the guide vanes, the entrance angle of the runner buckets, and the portion of the head h, which is converted into the velocity of entrance v''.

$$\phi = \frac{\pi Dn}{12 \times 60 \sqrt{2g}\sqrt{h}} = \frac{Dn}{1840\sqrt{h}} \tag{86}$$

The value of ϕ for reaction wheels may vary from

$\phi = 0$ with the turbine stationary, to

$\phi_{max.} = 1.15$ to 1.60 at the runaway speed, and

$\phi_e = .60$ to $.90$ with various types of turbines.

For tangential wheels:

$\phi_{max.} = .84$ to $.92$, and

$\phi_e = .43$ to $.48$

(88) $\triangle = 1840\,\phi = \dfrac{Dn}{\sqrt{h}}$ = r. p. m. of a wheel with $D = 1$ and $h = 1$ or the diameter of a wheel with $n = 1$ and $h = 1$.

$\triangle$, the speed coefficient = *the revolutions per minute of a one inch wheel under one foot head.* It is dependent on the same factors as ϕ, and its ordinary limits are therefore as follows:

For reaction turbines:

$\triangle_{max.} = 0$, with stationary turbines
$\triangle_{max.}$ = from 2116 to 2964
$\triangle_e$ = from 1124 to 1656

For tangential wheels:

$\triangle_{max.}$ = from 1546 to 1693
$\triangle_e$ = from 791 to 883

(96) $$q = c'a\sqrt{2gh}$$

c' is somewhat different from c as it involves an extra contraction due to the obstructive action of the bucket vanes (see Sec. 144). The area of the gate opening a, depends upon the height of the speed gate E, the diameter of the wheel D, and the angle of the guide vanes: hence for wheels of homologous design

(99) $$ca\sqrt{2g} = KD^2$$

with any fixed gate opening and consequent constant entrance angle; therefore

(100) $q = KD^2\sqrt{h}$, in which K varies with the same factors as a

$K = \dfrac{q}{D^2\sqrt{h}} = q$, with D and $h = 1$; and therefore

K = *Coefficient of discharge = the discharge in cubic feet per second of a one-inch wheel under one foot head.*

The limits of K depend on the design of the wheel and the gate opening and become O only with the gates closed.

(111) $$P = \frac{qhe}{8.8} = \frac{KeD^2h^{\frac{3}{2}}}{8.8} = \left(\frac{Ke}{8.8}\right)\left(D^2h^{\frac{3}{2}}\right)$$

Let $\dfrac{Ke}{8.8} = p$; then p varies with the discharge K, and efficiency e, of the wheel, and

(113) $P = pD^2h^{\frac{3}{2}}$ which transposed gives the expression for p, thus

$p = \dfrac{P}{D^2h^{\frac{3}{2}}} = P$ when $D = 1$ and $h = 1$; and therefore

p = *Coefficient of power and is defined as the power of a one-inch wheel under one foot head.*

$$\text{(117)} \qquad \mathfrak{P} = \Delta^2 p = \frac{n^2 P}{h^{\frac{5}{2}}}$$

$\mathfrak{P}$ is designated as the specific power coefficient and its value varies with the angle of the guide vanes, the entrance angle of the buckets, the entrance velocity of the water, the discharge of the turbine, and the efficiency of the wheel. With the first three factors, homologous the following relations obtain:

For tangential wheels:

$$\text{(129)} \qquad \mathfrak{P} = \frac{c\pi}{4} \cdot \frac{d^2}{D^2}$$

That is, $\mathfrak{P}$ is directly proportional to the ratio of

$$\frac{\text{the square of the diameter of the jet}}{\text{square of the diameter of the wheel}}$$

For reaction turbines:

$$\text{(131)} \qquad \mathfrak{P} = cc''' \frac{E}{D}$$

That is, $\mathfrak{P}$ is directly proportional to the ratio

$$\frac{\text{Height of speed gate}}{\text{diameter of wheel}}$$

(120) $N_u = \sqrt{\mathfrak{P}} = n\sqrt{\frac{P}{h^{\frac{5}{2}}}}$ = Unity speed = type characteristic, a coefficient used by turbine designers.

(122) $N_s = 4.46 N_u = 4.46\, n\sqrt{\frac{P}{h^{\frac{5}{2}}}}$ with English units.

(123) N_s = Specific speed $= n\sqrt{\frac{P}{h^{\frac{5}{2}}}}$ with metric units.

The value of these coefficients can always be determined for wheels of homologous design, when the head, speed diameter and discharge of the wheel are known from the results of experiment, by substitution in the formulas here summarized.

$$\phi = \frac{v'}{v} = \frac{Dn}{720\sqrt{2gh}} = \frac{.000543\, Dn}{\sqrt{h}}$$

$$\Delta = 1842\,\phi = \frac{Dn}{\sqrt{h}},$$

$$K = \frac{q}{D^2\sqrt{h}}$$

$$p = \frac{P}{D^2 h^{\frac{3}{2}}}$$

$$\mathfrak{P} = \frac{n^2 P}{h^{\frac{5}{2}}}$$

151. Variation in Turbine Coefficients.—For every condition of operation of a turbine corresponding to each combined condition of gate opening and speed relation ϕ or $\triangle$ there is a single value of K, p, e and P. If any wheel of a homologous series be tested, the corresponding values of the coefficients for each experimental determination can be platted, and in connection with other similar data, will furnish points on curves which will show graphically the way in which the characteristic coefficients vary under each condition of operation.

Figure 215, page 327, is a graphical log of the variations in the coefficients of a turbine under six conditions of gate opening.

This diagram contains four series of curves.

1. The efficiency-speed curves, showing the relation between e and ϕ or $\triangle$ for six conditions of gate. From these curves it will be noted that e varies from O at $\phi = o$ and $\triangle = o$ to a maximum at some particular value of ϕ or $\triangle$ which differs with each gate and becomes o again at a certain maximum value of ϕ or $\triangle$ which differs with each gate opening, $\phi_{max.}$ and $\triangle_{max.}$ occur at full gate; $e_{max.}$ occurs at .733 gate with $\phi_e = 81$ and $\triangle_e = 1490$.

2. The discharge-speed curves, showing the relation between K and ϕ or $\triangle$ for six conditions of gate. The discharge coefficient K varies with each gate and with the variation in speed ϕ and $\triangle$ but becomes zero only with zero gate openings.

3. The power-speed curves, showing the relation between the power coefficient p and ϕ or $\triangle$ for six conditions of gate. The power coefficient varies for each gate from o with ϕ and $\triangle = o$ to a maximum at certain values of ϕ and $\triangle$ which differ from those for $e_{max.}$ As the speed increases the power decreases to o at the same values of ϕ and $\triangle$ as in the case of the efficiency curves for the same gate.

4. The specific-power coefficient $\mathfrak{P}$ for full gate conditions is o with ϕ and $\triangle = o$; it reaches a maximum with $\phi = 1.03$ and $\triangle = 1895$. and again becomes o at $\phi = 1.27$ and $\triangle = 2347$, at which speed the efficiency and the power coefficient for full gate become o.

It will be noted that there is one maximum value of e in this diagram that corresponds to a certain value for each of the other coefficients, and the value of each coefficient which corresponds to this condition

for maximum efficiency is designated by a subscript e, i. e. ϕ_e, K_e, p_e and $\wp_e$.

In catalogs of standard turbines, the speed, power and discharge are usually based on these sub e values of the coefficient for the series, and

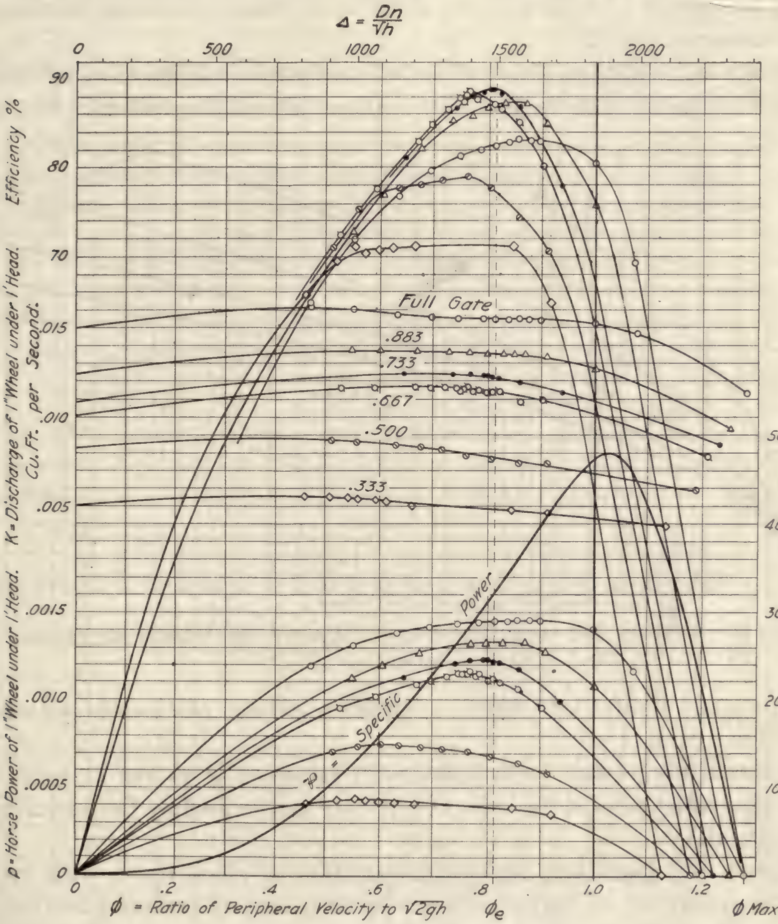

Fig. 215.—Graphical Log of the Variation in Turbine Coefficients.

these sub e values are also ordinarily used in giving a synopsis of the characteristics of turbines. As previously noted, it is not always desirable to run a turbine under these exact conditions, and for a thorough understanding in detail of the best operating conditions for a turbine under any given condition of load, a complete analysis of the turbine is essential.

152. Range of Turbine Design.—The range of variation which may be secured by turbines of different design should be understood and appreciated. Fig. 216, shows the relation of diameter to power in wheels of various designs, in which the power coefficient p, varies from $p = .001$ to $p = .005$. The latter is the maximum value that has been attained (1915).

Figure 217, page 329, shows the relation of power and speed of wheels of various specifi powers $\mathfrak{P}$. The diagram shows the present (1915)

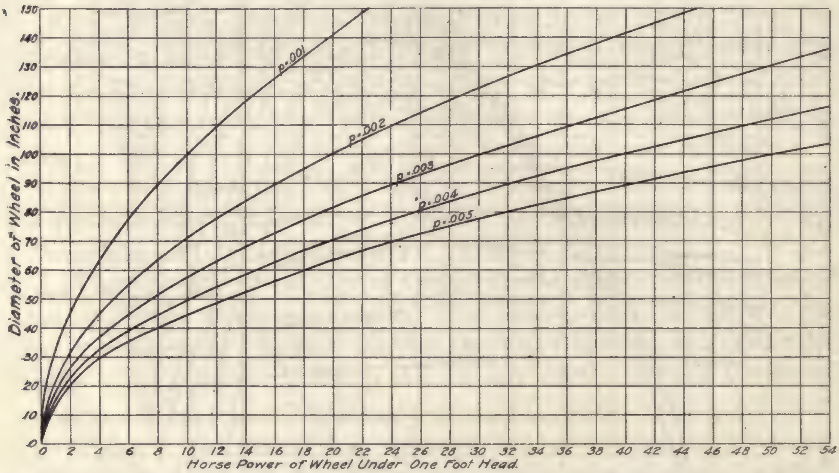

Fig. 216.—Relation Between Power and Diameter for Wheels of Various Design.

approximate limits for reaction turbines giving efficiencies above eighty per cent.

Reaction wheels may be designed with values of $\mathfrak{P}$ as low as 100.

In tangential wheels, the values of $\mathfrak{P}$ vary approximately from twelve to thirty-six.

Figure 218, page 330, shows the variation of efficiency e, with the specific power $\mathfrak{P}$, of various types of wheels and of various manufacturers. Each Roman number (I, II, etc.) indicates the tests of wheels of a single manufacturer, which tests are published in the Appendix (excepting IV which represents miscellaneous wheels, A being the wheel of one manufacturer, B that of another and C and D of a third).

The various values of turbine coefficients are no indication of the comparative value of certain designs of turbines, except as they apply to certain definite conditions under which they are to operate. When a certain definite power is to be developed at a given speed under a

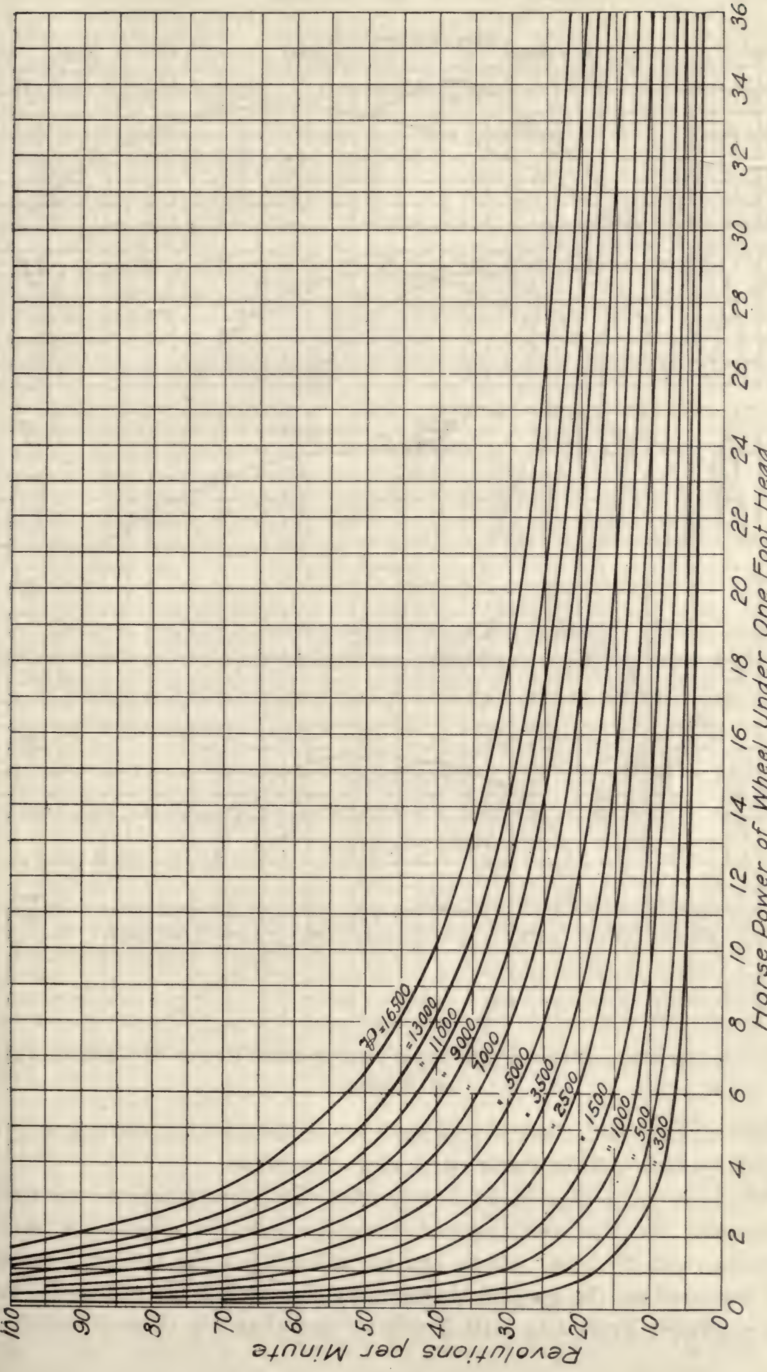

Fig. 217.—Relation Between Power and Speed for Wheels of Various Specific Powers.

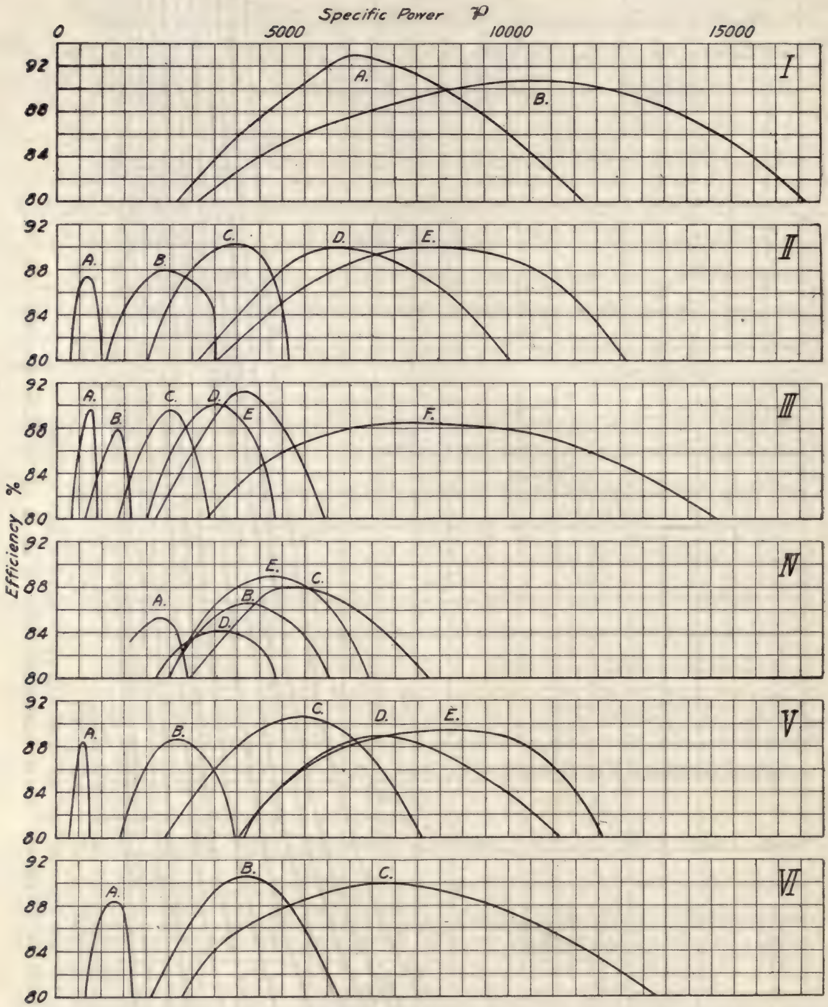

Fig. 218.—Relation Between Efficiency and Specific Power of Various Types of Wheels.

certain fixed head, these coefficients are indicative of the turbine or turbines which will fit those particular conditions.

When no particular speed is required, but a certain power is to be developed, the size and type of wheels which are obtainable under American practice and which can be utilized for the required service are indicated by the graphical diagrams shown in Fig. 216, page 328. For example: From Fig. 216, it will be seen that it is desired to deliver

640 H. P. under sixteen foot head, the corresponding power under one foot head will be ten H. P. (see equation 110; $P_1 = \frac{P}{h^{\frac{3}{2}}} = 10$). To develop this power under one foot head will require a wheel of the size shown in Table 32 depending upon the power coefficient.

TABLE 32.

Relation of Diameter to Power Coefficient for Wheels to Deliver Ten Horse Power Under One Foot Head.

Diameter of Wheel D Inches	Value of Power Coefficient p
100	.001
72	.002
58	.003
50	.004
45	.005

From Fig. 217, page 329, it will also be seen that for a turbine which will develop ten H. P. under one foot head (and consequently 640 H. P. under sixteen foot head) the relations as shown in Table 33 between specific power $\mathfrak{P}$, and sevolutions per minute will exist.

TABLE 33.

Relation of Revolutions to Specific Power for Wheels to Deliver Ten Horse Power Under One Foot Head.

Value of Specific Power	Revolutions per Minute	
	n_1	n_{16}
300	5	20
1000	9.8	39.2
2500	15.9	63.6
5000	22.5	90
9000	30	120
16500	40.6	162.4

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CHAPTER XII

TURBINE TESTING

153. **Symbols.**—The symbols and characters used in this chapter have the following significance:

E = Total energy available at the plant.

E_p, E_j, E_d, E_w, etc. = Energy available under some definite condition.

e = Efficiency.

g = Acceleration due to gravity = 32.2 feet per second per second.

π = Ratio of circumference to diameter of a circle = 3.1416.

ϕ = Ratio of spouting velocity of water to the peripheral velocity of the wheel.

h = Head of water acting.

h', h'', etc. = Head for some particular condition.

l = Length of lever or brake arm measured from the center of revolution in feet.

L = Length of weir crest in feet.

n = Revolutions per minute.

P = Horse power.

q = Quantity of discharge in cubic feet per second.

S = Weight on scale in pounds.

v = Velocity of flow in feet per second.

w = Weight of cubic foot of water 62.5 pounds.

154. **The Importance of Testing Machinery.**—A correct theory based on mathematical analysis forms a valuable foundation for machine design. In the construction of any machine, however, theoretical lines can seldom be followed in all details, and, even if this were possible, the truth of the theory must be demonstrated by actual trial for there are usually many factors involved which cannot be theoretically considered and yet affect practical results. In any machine much depends upon the character of the workmanship, on the class of material used, and on all the details of manufacture, installation and operation as well as on design. All of these matters can hardly be included in a theoretical consideration of the subject, and it therefore becomes necessary to determine the actual results attained by a trial of the machinery under working conditions.

General observations or even a detailed examination of any machine and its operation can rarely be made sufficiently complete to give any accurate knowledge of the quantity or quality of the results which it can and does accomplish. It is only when the actual effect of slight

changes in design can be accurately determined by careful experiment that a machine can be improved and practical or approximate perfection attained.

The ease with which such determination can be made is usually a criterion of the rapidity with which the improvements in the design and construction of a particular machine take place. Where such determinations are readily made, rapid advancement results, but where they are costly and require a considerable expenditure of time or money, the resulting delays and expenses usually so limit such determinations that good results are attained but slowly.

The invention of the steam engine indicator and the Prony brake placed in the hands of the engineer instruments by means of which he could readily determine the action of steam within the engine cylinder and the actual power developed therefrom. The knowledge thus gained has been one of the most potent factors in the rapid advancement of steam engineering.

The physical results of radical modifications or changes in design are sometimes quite different from those anticipated by the designer. Improvement in any machine means a departure from the tried field of experience and the adoption of new and untried devices or arrangements. Frequently a line of reasoning, while apparently rational, is found to be in error on account of unforeseen conditions or contingencies and the results anticipated are not borne out in the actual practical results. Unless, therefore, such results are carefully and accurately determined by exact methods the actual value of changes in design may never be known or appreciated and designs may be adopted which, while apparently giving a more desirable form of construction, actually accomplish less than the form from which the design has departed.

155. **The Testing of Water Wheels.**—The value of the testing of water wheels was recognized by Smeaton who tested various models of water wheels about the middle of the eighteenth century. Methods of turbine testing were also devised with the first development of the turbine, which have been potent factors in the improvement of the turbine. While the methods of testing have been greatly improved since that time, they have not as yet reached a state that can be considered reasonably satisfactory, and turbine testing has not become so general as to assure the high grade of design and workmanship in their manufacture as in other machinery where testing is more easily and regularly practiced.

The principal causes of the backward condition of turbine testing lie in the difficulties and expense of making an accurate test in place, and the expense and unsatisfactory results of testing turbines in a testing flume where the head and capacity are so limited as to confine satisfactory tests to heads of seventeen feet or less and to wheels of a capacity of about 250 cubic feet per second, or less if the full head of seventeen feet is to be maintained. There is an urgent demand for accurate and economical methods for the measurement of the water used and of the power developed by water wheels in place, that can be readily and quickly applied without the almost prohibitive expense of the construction of expensive weirs and other apparatus now used for such purposes. Apparently slight variations in turbine construction produce radical changes in practical results. The efficient results achieved under test by a well-designed and well-constructed wheel is no assurance that wheels of the same make and of the same design, even though they be of the same size and even from the same pattern, will give similar results when set in place with different casings and under practical operating conditions. This is especially true when the contingencies of competition and the knowledge that a test of the wheel is impossible, or at least highly improbable, offer a premium on careless construction and cheap work.

A brief examination of the work already done in this line, and of the methods now in vogue, may afford suggestions for future improvements and development in this important work.

156. Smeaton's Experiments.—John Smeaton, the most experienced and eminent engineer of his time, made a series of experiments on the power and effect of water used by means of various forms of water wheels for mill purposes. Accounts of these experiments were published in the Transactions of The Royal Society of England in 1759. Until that time the relative values of the different types of water wheels of that day were very poorly understood and appreciated.

Smeaton's apparatus for measurement of the power of overshot and undershot wheels is shown by Figs. 219, 220, pp. 337, 338.* Water was pumped by means of the hand pumps from the tail basin *X*, to the supply cistern *V*, from which it was admitted to the wheel through an adjustable gate. The power developed was measured by the time required to raise a known weight through a known height by means of a cord passing through a system of pulleys and attached to a small winding drum or collar upon the wheel shaft. This drum revolved only

* The Encyclopedia of Civil Engineering, by Edward Cressy.

when, by slight longitudinal movement, it was made to engage a pin on the shaft.

In these experiments Smeaton found a maximum efficiency of thirty-two per cent., and a minimum efficiency of twenty-eight per cent. for undershot wheels. He also observed that the most efficient relations between the peripheral velocity of the wheel and velocity of the water were attained when the former was from fifty per cent. to sixty per cent. of the latter, and that the force that could be exerted by a wheel to advantage was from fifty per cent. to seventy per cent. of the force required to maintain it in stationary equilibrium.

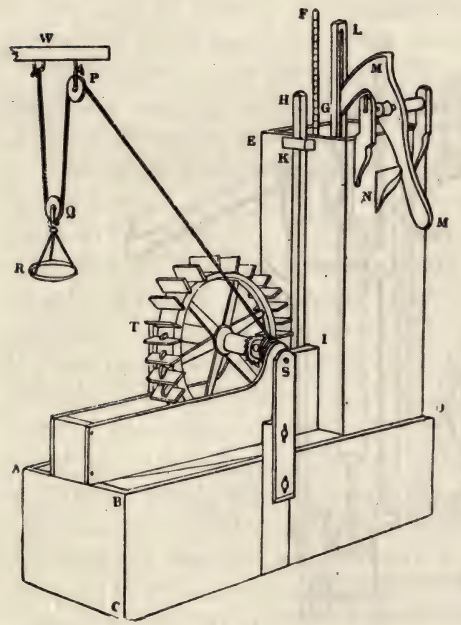

Fig. 219.—Smeaton's Apparatus for Testing Undershot Water Wheels.

For overshot wheels Smeaton found that the efficiency varied between fifty-two and seventy-six per cent. From his experiments he concluded that the overshot wheel should be as large as possible, allowing, however, a sufficient fall to admit the water onto the wheel with a velocity slightly greater than that of the circumference of the wheel itself, and that the best velocity of the circumference of the wheel was about three and one-half feet per second. This speed he found applied both to the largest as well as to the smallest water wheel.

From these experiments Smeaton concluded that the power of water applied directly through the exertion of its weight by gravity, as with the overshot wheel, was more effective than when its power was applied through its acquired momentum, as in the undershot wheel, although his line of reasoning indicated otherwise. The later development of impulse wheels shows that his reasoning was correct, and that the low efficiency of the impulse wheel was due to the method of applying the momentum of the water rather than to any inherent defect in the impulse principle.

The experiments or tests of Smeaton, while crude and imperfect and performed upon wheels which were merely models, afforded a comparative measurement of the efficiency of the undershot, overshot and breast wheels then in use and had a marked effect on the further selection of such wheels.

157. The Early Testing of Turbine Water Wheels.—The testing of turbine wheels began many years ago in France before the turbine became well known in the United States.*

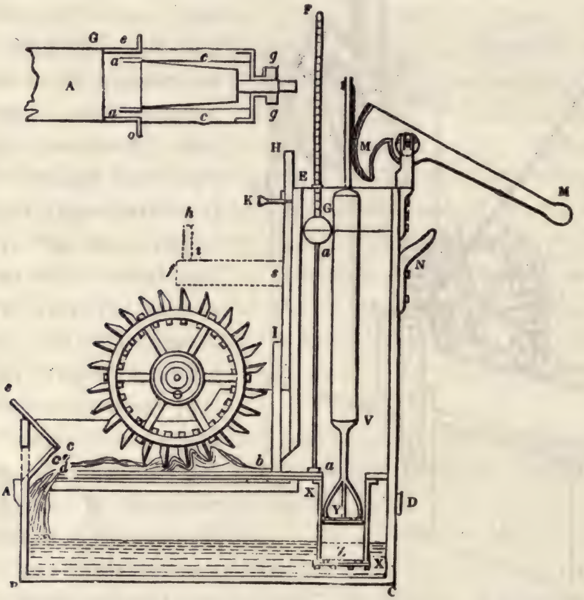

Fig. 220.—Section of Smeaton's Apparatus for Testing Water Wheels (see page 336).

Fourneyron began the study of the early forms of turbines as early as 1823, and, in 1827, he introduced his well-known wheel and also brought into notice a method of systematic testing of the same by means of the Prony brake.

"La Societé d' Encouragement pour l' Industrie Nationale" is credited by Thurston with the introduction of a general system for the comparison of wheels and correct methods of determining the efficiency.† Other engineers immediately accepted this method of com-

* See "The Systematic Testing of Water Wheels in the United States," by R. H. Thurston, Trans. Am. Soc. Mech. Eng. vol. 8.

† See "Memoire sur les Turbines Hydrauliques," by R. Fourneyron, Brussels, 1840.

parison of wheels. Morin, in 1838, reported the results of a trial of a Fourneyron wheel as giving an efficiency of sixty-nine per cent. with only slight changes in values for a wide range of speed. With another wheel he obtained seventy-five per cent. efficiency.*

Combes tested his reaction wheel and found that an efficiency of about fifty per cent. could be obtained.†

The first systematic test of turbines in the United States was made by Mr. Elwood Morris of Philadelphia in 1843 and reported in the Journal of The Franklin Institute for December of that year.

The maximum efficiency reported was seventy-five per cent. This result was reached when the value of ϕ for the interior circumference of the Fourneyron turbine was .45. In 1844 Mr. James B. Francis determined the power and efficiency of a high breast water wheel in the city of Lowell, using a Prony brake fitted with a dash-pot to prevent irregular operation.

In 1845 Mr. Uriah A. Boyden made a trial of a turbine designed by himself, using the Prony brake, and obtained an efficiency of seventy-eight per cent. as the maximum. In 1846 a similar test of one of the Boyden turbines was made at the Appleton Mills in Lowell, and an efficiency of eighty-eight per cent. was reported. He continued the work of the testing of water wheels for several years and tested many wheels of various types.‡ Mr. Francis introduced the system of testing wheels which were to be used by purchasers of water from the water power company which he represented. The chief purpose of the tests was that the wheels might be used as meters in determining the amount of water used by the various purchasers.

In 1860 the City of Philadelphia undertook a comparative trial of various turbines in order to determine their relative merits for use in the Fairmount Pumping Plant. The results of these tests given in Table 34 are somewhat questionable but have a comparative value.

158. **The Testing of Turbines by James Emerson.**—One of the men who did much valuable work of this character was Mr. James Emerson who designed a new form of dynamometer of the transmitting kind. At the request of Mr. A. M. Swain, Mr. Emerson designed a Prony brake, embodying this dynamometer for the purpose of testing a Swain turbine in a flume built from designs by Francis. The re-

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* See "Experiences sur les Power Hydrauliques," Paris, 1838.

† See "Mechanics of Engineering," Weisbach. Translated by A. J. DuBois, Hydraulics and Hydraulic Motors, vol. II, part I, p. 470.

‡ See "Lowell Hydraulic Experiments."

sults obtained by Mr. Emerson from this test were so satisfactory that The Swain Turbine Company decided to open the flume for the purpose of a competitive test of all turbines which might be offered for this purpose. Announcement of this test was dated June 16, 1869.

TABLE 34.

Water Wheel Tests at Philadelphia in 1860.

Name of Wheel.	Kind of Wheel.	Per cent of Effect	3 per cent added for frict'n	Where built.
Stevenson's second wheel....	Jonval..	.8777	.9077	Paterson, N. J.
Geyelin's second wheel......	Jonval..	.8210	.8510	Philadelphia, Pa.
Andrews & Kalbach's third wheel	Spiral...	.8197	.8497	Bernville, Pa.
Collin's second wheel........	Jonval..	.7672	.7972	Troy, N. Y.
Andrews & Kalbach's second wheel	Spiral...	.7591	.7891	Bernville, Pa.
Smith's, Parker's fourth trial	Spiral...	.7569	.7869	Reading, Pa.
Smith's, Parker's third trial.	Spiral...	.7467	.7767	Reading, Pa.
Steven's first wheel.........	Jonval..	.7335	.7635	Paterson, N. J.
Blake	Scroll...	.7169	.7469	East Pepperell, Mass.
Tyler	Scroll...	.7123	.7423	West Lebanon, N. H.
Geyelin' first wheel.........	Jonval..	.6799	.7099	Philadelphia, Pa.
Smith's, Parker's second wheel	Spiral...	.6726	.7026	Reading, Pa.
Merchant's Goodwin..........	Scroll...	.6412	.6712	Guilford, N. Y.
Mason's Smith...............	Scroll...	.6324	.6624	Buffalo, N. Y.
Andrew's first wheel........	Spiral...	.6205	.6505	Bernville, Pa.
Rich	Scroll...	.6132	.6432	Salmon River, N. Y.
Littlepage	Spiral...	.5415	.5715	Austin, Texas.
Monroe	Scroll...	.5359	.5659	Worcester, Mass.
Collin's first wheel..........	Jonval..	.4734	.5034	Troy, N. Y.

The pit was fourteen feet wide, thirty feet long, and three feet deep, measured from the crest of the weir. The best results of this competitive test, the accuracy of which has since been questioned by Mr. Emerson, were attained with the Swain and Leffel wheels. The former ranged from 66.8 up to 78.9 per cent. efficiency, and the latter from 61.9 to 79.9 per cent. efficiency. This competitive test was the beginning of a series of such tests as well as of a general system of the public testing of turbines. The testing flume was opened to all builders and users of turbine wheels and such tests have been continued in the United States up to the present time.

The report of the results of this test attracted the attention of Mr. Stewart Chase, then agent of The Holyoke Water Power Company,

who, recognizing its very great importance, secured the adoption of a systematic testing of water wheels at Holyoke for the benefit of the company and wrote to Mr. Emerson as follows:

"The testing of turbines is the only way to perfection, and that is a matter of great importance. Move your work to Holyoke and use all the water that is necessary for the purpose, and welcome, free of charge."

Mr. Emerson, who had been conducting the testing of water wheels as a matter of private business at Lowell, at which place he was obliged to pay for the water used, at once accepted the liberal offer thus tendered him and removed to Holyoke where he continued the testing of water wheels until it was taken in hand by The Holyoke Water Power Company.

The reports of Mr. Emerson's work were published and undoubtedly were the means of bringing a number of wheels up to a state of high efficiency. The reports were found to be full of valuable data, and, although not systematically arranged, formed an extensive and valuable collection of figures.*

In 1879, The Holyoke Water Power Company, for the purpose of determining the standing of wheels offered for use at that place, arranged for a comparative or competitive turbine test at the flume constructed by Mr. Emerson at Holyoke. The wheels were set under the direction of Mr. Emerson and a part of the tests were made or witnessed by Mr. Samuel Webber and Mr. T. G. Ellis. Their report was accompanied by a graphical diagram reproduced in Fig. 221, page 342.

The report of Mr. Emerson covered a much larger number of wheels. The diagram accompanying Mr. Emerson's report * is reproduced in Fig. 222, page 343.

159. The Holyoke Testing Flume.—The later work of systematic testing of American turbines has been carried on principally at the Holyoke flume.

‡ "The object aimed at by the Water-power Companies of Lowell and Holyoke, in the establishment of testing flumes for turbines, is the determination of the power and efficiency, the best speed, and the quantity of water flowing at from whole, to, say, half gate, so exactly that the wheel may be used as a meter in the measurement of the water used by it. The quantity of water passing through the wheel, at any

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* See James Emerson's "Hydro-Dynamics."

† Emerson's "Hydro-Dynamics," page 300.

‡ "The Systematic Testing of Water Wheels," by R. H. Thurston.

given gate-opening, will always be practically the same at the same head, and the wheel having been tested in the pit of the testing flume, and its best speeds and highest efficiency determined, and a record

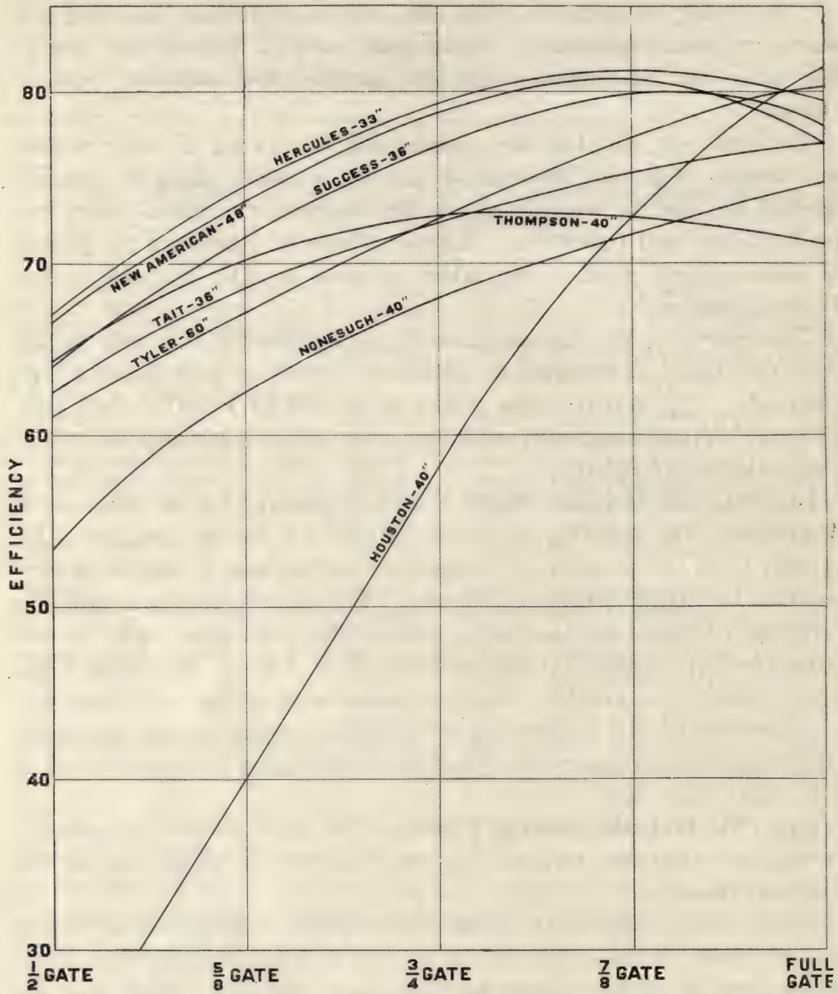

Fig. 221.—Webber and Ellis' Results of Tests of Water Wheels (1879) (see page 341).

having been made of the quantity of water discharged by it at these best speeds and at all gates, the turbine is set in its place at the mill, speeded correctly for the head there afforded, and a gauge affixed to its gate to indicate the extent of gate opening. The volume of water

passing the wheel at various openings of gate having been determined at the testing flume, and tabulated, the engineer of the water power company has only to take a look at the gauge on the gate, at

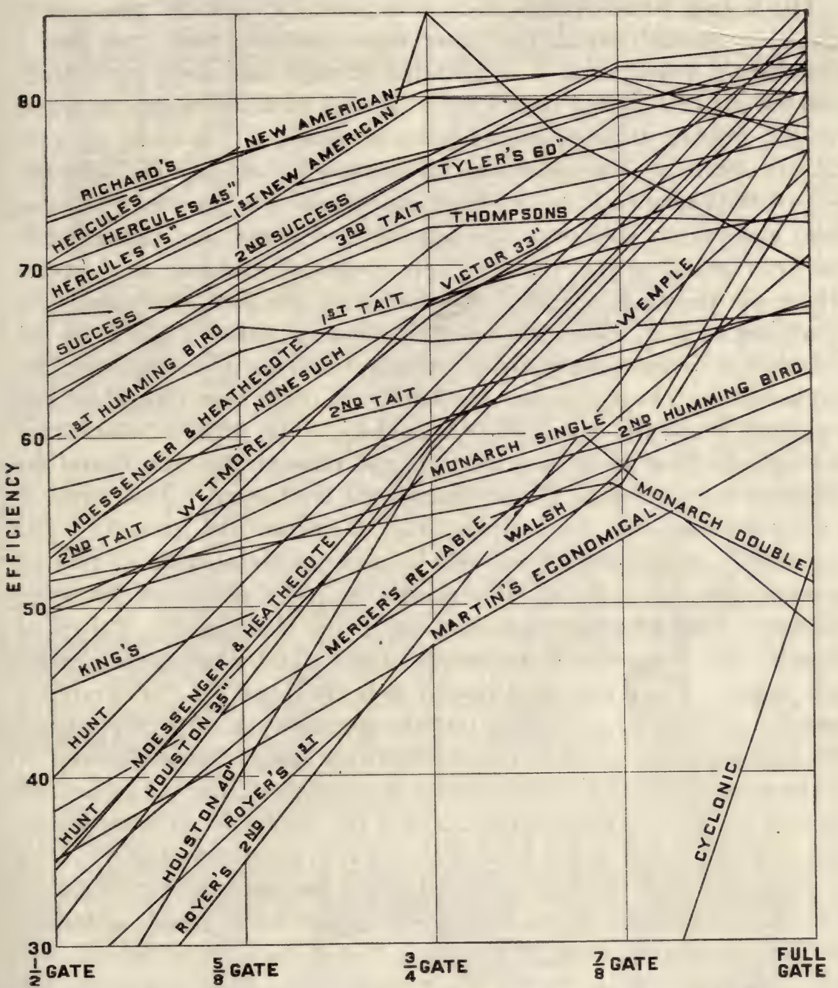

Fig. 222.—Results of Emerson's Tests of Reaction Turbines (1879) (see page 341).

any time, or at regular times, and to compare its reading with the table of discharges, to ascertain what amount of water the wheel is taking and to determine what is due the company for the operation of that wheel, at that time. The wheel is thus made the best possible meter for the purposes of the vender of water."

The present Holyoke testing flume was completed in 1883. A plan and section of this flume are shown in Figs. 223, page 345 and 224, page 346.

The testing flume consists of an iron penstock *A,* about nine feet in diameter, through which the water flows from the head race into a chamber *B,* from which it is admitted through two head gates *G, G,* into the chamber *C,* and from thence through trash racks into the wheel pit *D.* Passing through the wheel to be tested, it flows into the tail-race *E,* where it is measured as it flows over a weir, at *O.* The object of the chamber *B,* is to afford opportunity for the use of the two head gates *G, G,* to control the admission of water, and consequently the head acting on the wheel. There is also a head gate at the point where the penstock *A,* takes in water from the first level canal. A small penstock *F,* about three feet in diameter, takes water from the chamber *B,* independently of the gates and leads to a turbine wheel *H,* set in an iron casing, in the chamber *C,* in order that this wheel can run when *C* and the wheel pit *D,* are empty. The wheel *H,* discharges through the floor at the bottom of *C,* and through the arch *I,* and the supplementary tail-race *K,* into the second level canal. This wheel is used to operate the repair shops; also to operate the gates *G.* The chamber *C,* is bounded on one side by a tier of stop-planks *L,* and, on another side, by a tier of stop-planks *M.* The object of the stop-planks *L,* is to afford a waste-way out of the chamber *C.* This is of especial use in regulating the height of the water when testing under low heads. The water thus passed over the planks *L,* falls directly into the tail-race *K,* and passes into the second level. The stop-planks *M,* are used when scroll or cased wheels are tested. In such cases *D* is empty of water and the wheel case in question is attached by a short pipe or penstock from an opening cut in the planks *M.* Flume wheels are set in the center of the floor of *D,* and *D* is filled with water. They discharge through the floor of *D* and out of the three culverts *N, N, N,* into the tail-race *E.* Horizontal wheels are set in the pit *D,* with their shafting projecting through a stuffing-box in the side of the pit (see Fig. 224, page 346). At the down-stream end of the tail-race is the measuring weir *O* (Fig. 223, page 345). The crest of the weir is formed of a strip of planed iron plate twenty feet in length. The depth of water on the weir is measured in a cylinder *P,* set in a recess *Q.* fashioned in the sides of the tail-race. These recesses are water-tight, and the observer is thus enabled to stand with the water-level at convenient height for accurate observation. The cylinder *P,* is connected

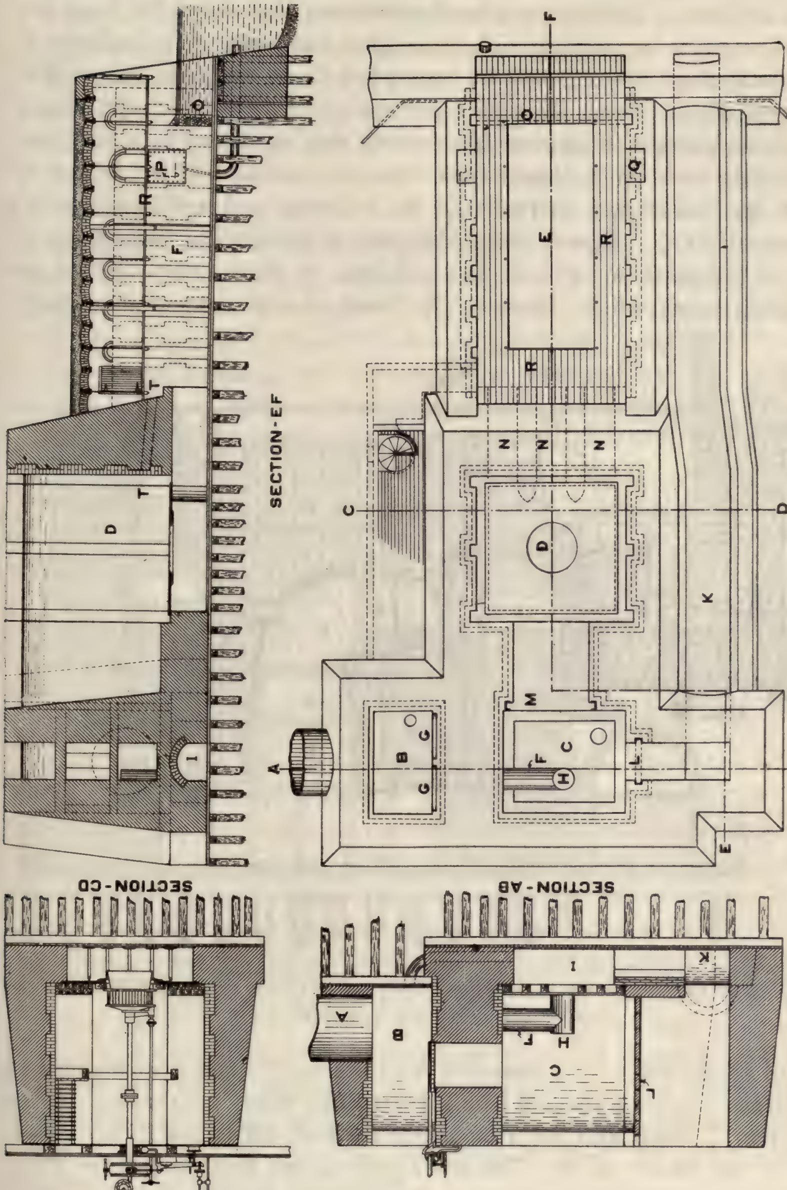

Fig. 223.—Testing Flume of Holyoke Water Power Co. (see page 341).

with a pipe that crosses the tail-race or weir box about ten feet back of the weir crest. The pipe is placed about one foot above the floor and is perforated in the bottom with one-eighth inch holes. A platform *R*, surrounds the tail-race, and is suspended from the iron beams that carry the roof. Above the tail-race is the street, over which the wheels to be tested arrive on wagons from which they are lifted by a traveling crane that runs on a frame-work over the street, and by means of which the wheels are carried into the building and are lowered into the wheel pit *D*. Spiral stairs lead into a passageway that leads in turn to the platform *R*. In the well-hole of these stairs are set up the glass tubes which measure the head of water upon the wheel.

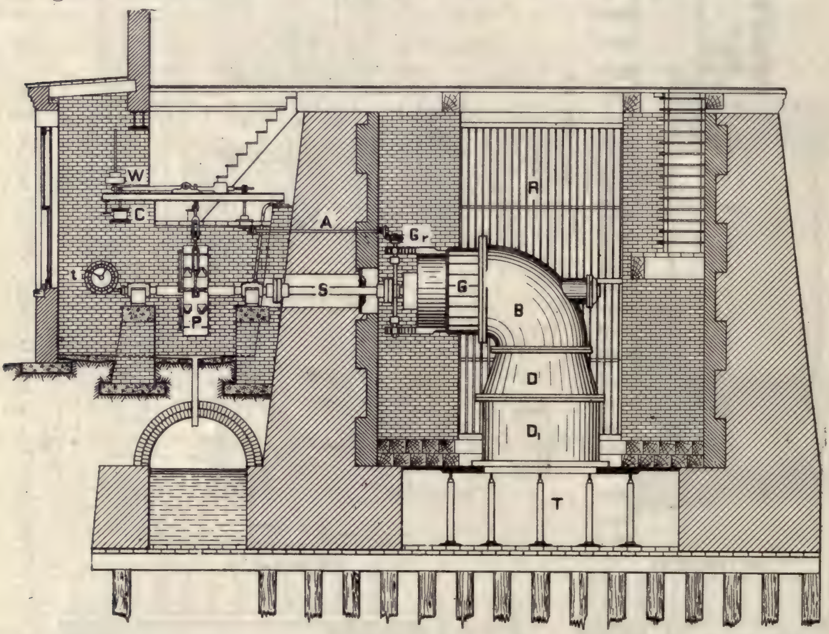

Fig. 224.—Testing Flume of Holyoke Water Power Co. Arranged for Testing Horizontal Turbines (see page 341).

These gauge tubes are connected with the pit *D*, and the chamber *C*, by means of pipes, one of which enters the wheel pit through a cast iron pipe *T*, built into the masonry dam which forms the down stream end of the wheel pit *D*. The other pipe passes back under the wheel pit *D*, and crosses the tail-race at the extreme back line and close under the pit floor. This pipe is perforated throughout its length across the race in a manner similar to the pipe used for determining the head

on the weir. To enable the observers at the brake wheel, head gauge and measuring weir to take simultaneous observations, an electric clock rings three bells, simultaneously, at intervals of one minute.

The usual method of testing a wheel is as follows: After the wheel is set in place (see Figs. 223 and 224) a brake pulley and Prony brake are attached to the shaft, the gates are set at a fixed opening and water is admitted. The runaway speed of the wheel is first determined with the brake band loose, after which a weight is applied and the brake tightened until the friction load balances the weight. As soon as this balance is attained, which requires only a few seconds, the revolution counter is read and the heads in the head-race, tail-race and on the weir are observed. Observations are repeated simultaneously each minute at the stroke of the bell and for a period of from three to five minutes. The weight is then changed and the observations repeated for a different load and speed. After observations are made over the range of speeds desired, the gate opening is changed, and a similar series of observations are made for the new gate opening. This is repeated for each desired gate opening, usually from full gate to about one-half gate.

The results are calculated and reported in the form shown in Table 61 et seq., Appendix B. It is usually stated in the report whether the test is made with a plain or conical draft tube, whether plain or ball bearings are used, and also the pull necessary, at a given leverage, to start the turbines in the empty pit. No attempt is made in these reports to describe the bearings or finish of the wheels in detail.

The maximum head available is about seventeen feet under small discharges and this decreases to about nine feet under a discharge of 300 cubic feet per second. The capacity of the tail-race and weir is hardly sufficient for the accurate measurement of the latter quantity.

160. The Value of Tests.—There can be no question as to the very great value of carefully-made tests of any machine. It must be borne in mind, however, that any test so made represents results under the exact conditions of the test, and, in order to duplicate the results, the conditions under which the test was made must be duplicated. Any changes in the design or finish of the wheels, any alterations in the method of setting, or in the gates, draft tube or other appurtenances connected with the same are bound to affect the power and efficiency to a greater or less extent.

It is unfortunate for the world's progress that the records and conditions of failures are seldom made known. The record of a failure,

while of great value from an educational standpoint, may considerably injure the reputation of an engineer or manufacturer, and consequently results of tests and experiments, unless fully satisfactory, are seldom published or known except by those closely interested. For this reason, the published tests of water wheels usually represent the most successful work of the maker and the best practical results he has been able to secure. Tests, unless fairly representative, do not assure that similar turbines of the same make, or even similar turbines of the same make, size and pattern, will give the same efficient results unless all details of their design, construction, and installation are duplicated. There is no doubt that in many cases the published tests of water wheels are the final consummation of a long series of experiments, made in order to secure high results, and do not give assurance that such results can be easily duplicated. The manufacturers of standard wheels have acknowledged this by calculating their standard tables on a basis of power and efficiency below that of the best tests they are able to obtain, and it is only a matter of reasonable precaution for the engineer, who is utilizing the results of any such tests for the purposes of his design, to discount the test values to such an extent as will assure him that his estimates will be fulfilled.

In water wheels for high-grade work, it is important that the specifications for their construction be carefully drawn and that, by inspection and tests, the results of the work be fully assured. It is unfortunate that no easily applied method is available for the testing of water wheels in place. Such tests as are now carried on involve the shipment of one or more of the wheels from the place of manufacture to Holyoke, their tests under the conditions there obtainable and their reshipment to the point where they are to be installed. Here they may be set to operate under conditions entirely different from those of the Holyoke test and the actual results obtained cannot be easily determined. The most desirable test of any machine is a test made under the conditions of actual service, and, when such tests can be made, the results are much more definite and of greater value than the tests of the wheel made under conditions entirely different from those under which it is to operate.

The Holyoke testing flume is performing valuable service and the results of its work have been of material assistance in the development and improvement of a number of high grade wheels. Much remains to be done, however, in the development of turbine testing so as to make it possible to more readily determine results under working con-

ditions. More uniform work will undoubtedly result as the mechanical methods of manufacture improve and manufacturers are able to more nearly duplicate the satisfactory conditions which they have found to obtain in special cases.

161. Purpose of Turbine Testing.—Water turbines may be tested for various purposes among which may be named:

First: To determine the relations of discharge, power, head, speed, and efficiency, and the range of satisfactory operation of a particular turbine of fixed size and design.

Second: To determine coefficients or constants of power, discharge, and efficiency of a turbine of special design under various conditions of head and speed, as a basis for establishing the range of satisfactory operation and the most favorable conditions for a type or series of turbines of homologous design.

Third: To determine the power, discharge, and efficiency of a given turbine set in place, under a given speed and head, and whether it will fulfill the guarantees of its manufacturer.

Fourth: To investigate the effects of modification in design, and the various losses in the turbine, as a basis for the reduction of such losses by proper design.

The quantities to be measured in a water wheel test are head, revolutions per minute or speed, discharge, and power output.

In order to give a full knowledge of the evident range of satisfactory operating conditions of a turbine, it is important that observations be made at runaway speed and under stationary conditions for each gate opening. This should be done in order to determine the relations of resistance and speed for the full range of operation. In most Holyoke tests, numerous measurements are made over the speeds of maximum power and efficiency; but the extreme points should be taken even at the sacrifice of some of the other observations, as the extremes together with the remaining point near the most efficient speed will give broader and more important information, and serve as a most important check on the correction and consistency of the results.

Wherever possible, turbines to be tested at Holyoke should be set up with the chute cases, gates and guides with which they are designed to be used, as a change in these details may make a marked difference in the final performance of the wheels. In case a different setting is used in the final installation, a difference in the results will be due to the change in setting. For the first and second purposes above mentioned the range of experiments should be as complete as practicable,

the discharge and power of the wheel being determined from numerous speeds from a stationary condition to the runaway speed and at various heads within the limits of the physical conditions.

For the third purpose, the test may be carried only through the range of commercial conditions under which the wheel would ordinarily operate, but should, however, be broad enough to include the range of conditions which will obtain in practice due to the variations in head which are anticipated at various seasons.

In the fourth case the determination of heads, velocities and friction losses at various points in the wheel case and wheel may be essential. For this purpose a special line of investigation and tests are necessary, which, while of great importance, are of special interest to the manufacturer only or to those interested in the detail development of some wheel for special purposes.

For the purpose of any test a clear conception of the nature of the information sought is essential and each determination must be made with proper precaution in order to secure accurate results.

162. Factors that Influence the Results of a Test.—It is apparent from the principles discussed in Chapter II that the actual power developed by a turbine will be somewhat less than the theoretical power of the water passing into it, depending on the character of the wheel and the various energy losses involved in its operation. The efficiency of the wheel, representing the ratio of power developed to power applied, depends on the same factors.

These losses, incidental to the operation of a turbine, include the friction of the shaft on its bearings, the hydraulic resistance from the friction and shock of the water in the guides and passages, the slip or leakage between the fixed and revolving parts of the wheel, and the unutilized energy due to the velocity remaining in the water when discharged from the wheel.

These losses in modern turbines are estimated as follows: *

Shaft friction....................................	from 2 to 3 per cent.
Slip or leakage....................................	from 2 to 3 per cent.
Hydraulic friction and shock....................	from 10 to 15 per cent.
Energy in water leaving wheel..............	from 3 to 7 per cent.
Total loss of energy....................	from 17 to 28 per cent.

* See also "Development of Transmission of Power," by Unwin, p. 104; and "Francis Turbinen," by Muller, p. 18.

The total losses given above correspond to the best current practice. Under the best conditions efficiencies greater than ninety per cent. are now (1915) often obtained, and, under unfavorable conditions, with poor design and poor construction, efficiencies much less than the minimum are common. While the losses can never be eliminated they should be reduced to the practical minimum that good design and good workmanship will permit.

163. **Measurement of Discharge.**—The discharge q, of the wheel is commonly measured in cubic feet per second and must include only the actual discharge through the wheel itself. All leakage around the wheel or by the point of measurement, must be obviated or determined and deducted from the estimated turbine discharge, or added to the amount measured, as the case may be. The methods of measuring discharge include the standard weir, the submerged weir and orifice, the current meter, floats, moving screen, Venturi meter, color solution, salt solution, and the turbine itself as a meter. The relative merits of these various methods of measuring water depend both on the favorable conditions of the locality and the care and experience of the engineer in charge. All methods are subject to large errors, if carelessly applied.

The Weir and Orifice.—The standard weir is the basis for all discharge measurements at the Holyoke testing flume, and when properly applied, is perhaps the most satisfactory method. The actual weir coefficient must be known either by a direct calibration of the weir or by the construction of the weir on standard lines, that is on lines for which the discharge coefficients are well established. Standard weirs, properly used, should give results correct within one or one and a half per cent. Errors in weir measurements often reach values of five per cent. or more, due to the erroneous use of coefficients obtained from other weirs not strictly comparative, or in the measurement of head.

The head on the weir must be accurately measured in a proper stilling chamber connected to the weir channel some distance up stream from the crest and by means of a hook guage which should usually read to 1/1000 of a foot. An error of .01 foot in reading the head on the weir represents about one per cent., and an error of .001 about one-tenth per cent., in the computed discharge with a one and five-tenths head on the weir, and a much greater error at a lower head.

The construction of weirs in the tail-race of power plants, especially where large quantities of water are used under low heads, involves an

expense which is often prohibitive. In addition to this, the construction of such weirs in plants working under low heads would often seriously reduce the head and may so alter the working conditions as to render the test of little value.

Submerged Weirs and Large Submerged Orifices, which do not involve the loss of head necessary in the standard weir, would seem to offer methods of measurement worthy of consideration. The difficulties of measuring head, the larger percentage of error due to errors in measuring head, and the limited information concerning coefficients of discharge, has not led to any considerable use of this method. Head gates may be so constructed as to form standard submerged orifices, when partially closed, which on further investigation and experiment may provide a satisfactory method for the measurement of the discharge of turbine in place.

Current Meters, when properly calibrated and carefully used under good hydraulic conditions, and by a skilled observer, will give results closely approximating the weir in accuracy. In turbulent waters, the enclosed type of screen meter is found to be more accurate.

Moving Screens.—A moving screen, which practically fills the cross-section of a uniform channel leading to or from the wheel, has been used for some time in Europe and is now being introduced into this country. It is usually expensive of installation but involves no loss of head and is regarded as one of the most accurate methods of flow measurement.

Floats.—With a uniform channel of considerable length and good hydraulic conditions, float measurements properly made are very satisfactory. This method can be applied only under rather special and unusual conditions.

Color Solution.—Mr. R. D. Johnson recently measured the rate of water consumption in certain large turbines under a high head (240 feet) by injecting coloring matter into the head of the conduit and timing its appearance in the tail-race, thus determining with considerable accuracy the velocity in the penstocks and hence the rate of discharge.

Salt Solution.—This method consists of discharging a concentrated salt solution at a definite rate into the head waters in such a manner that it will become perfectly mixed with the stream, and by accurately analyzing the amount of salt found in the discharge from the wheels. Uniform application, precise analyses both before and after discharge, and perfect mixing, are the conditions necessary for accuracy in re-

sults. Mr. B. F. Groat believes that by this method the discharge of the turbines at Messena, New York, was determined "with a margin of error of only a few hundredths of one per cent."

Pitot Tubes.—Pitot tubes are especially adapted to use under high velocities in closed penstocks. The correct use of a properly calibrated Pitot tube, in long straight reaches of pipe, will give entirely satisfactory results. Its use below bends or obstructions must be accompanied by a traverse of the entire cross-section of the pipe on at least two diameters ninety degrees apart, and under every condition of flow; otherwise the position of the point of mean velocity cannot be correctly determined.

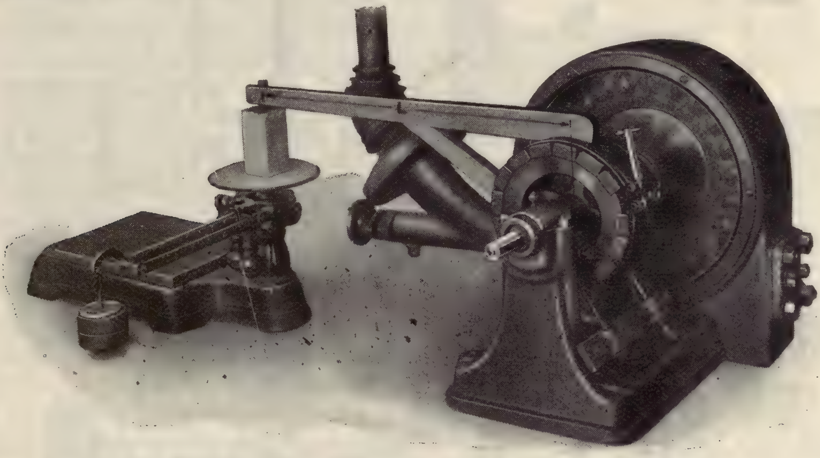

Fig. 225.—Doble Tangential Wheel Arranged for Brake Test **(see page 355).**

Venturi Meters.—Venturi meters have been installed in a number of plants supplied through closed penstocks. This provides a method of measurement for permanent use which may be considered as thoroughly reliable.

The Turbine as a Meter.—A turbine listed at Holyoke in its own chute case, with its permanent guides and gates, constitutes a good meter in itself. After installation, it may be tested for power either by means of a direct connected generator or by a suitable brake, and if the power developed at various gate openings substantiates the Holyoke tests, the results may usually be regarded as fairly satisfactory. Allen states that from several hundred tests of wheels in place he has found this to be true "within commercial requirements."

It is desirable with most systems of testing to check one method by another, if practicable.

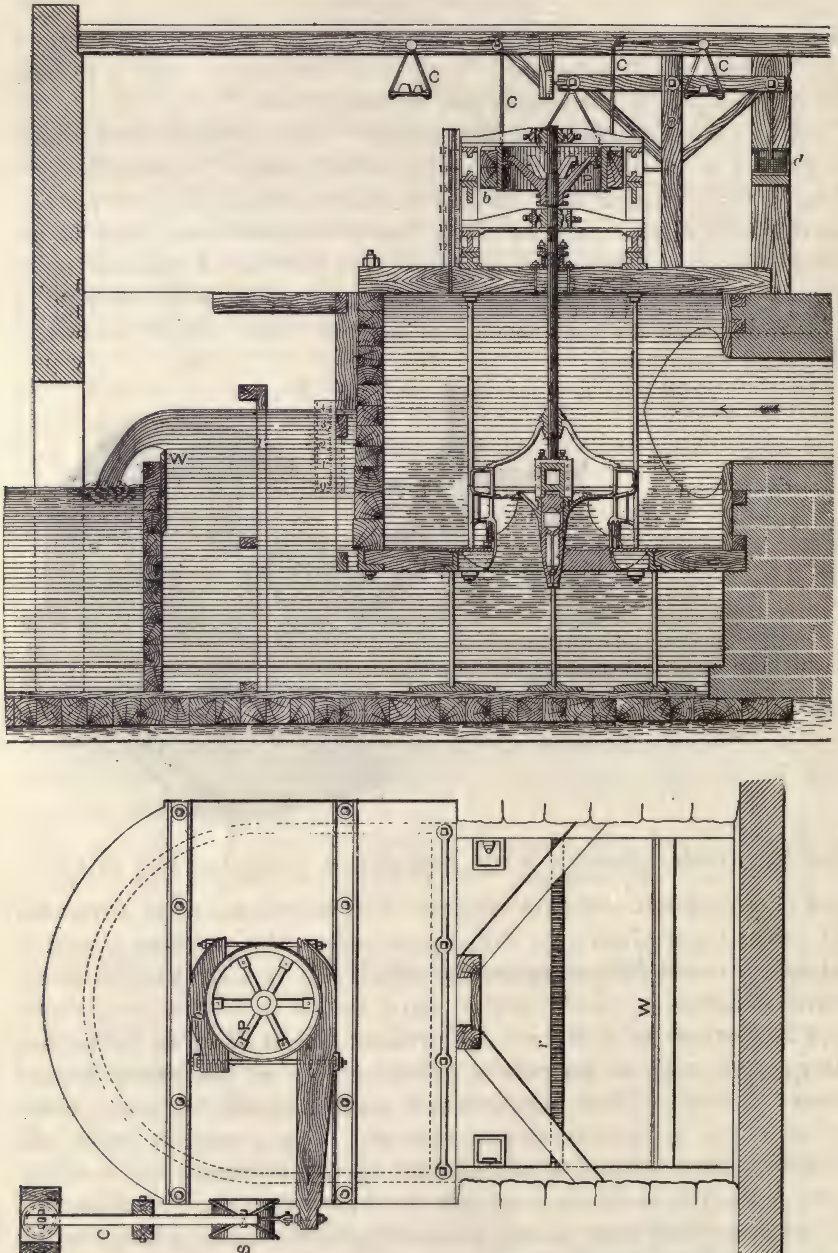

Fig. 226.—Section and Plan of Apparatus for Testing Swain Turbine (by James B. Francis). (See page 355.)

164. **Measurement of Head.**—The power of water applied to the wheel depends on both quantity and head. The head is more easily measured than the quantity, but, nevertheless, requires considerable care for its acccurate determination.

The head on the wheel must be measured immediately at the wheel both for the head-water and tail-water. If measured some distance away it is apt to include friction losses, which should not be charged against the wheel in raceways, penstocks and gates. The measurement of head should usually be to about .01 feet, although this depends on the magnitude of the heads involved.

165. **Measurement of Speed of Rotation.**—The speed of the wheel is usually recorded in revolutions per minute and may be determined by a revolution-counter which records the number of revolutions made in a given interval of time; or by a "tachometer" which, by means of certain mechanism, indicates at once on a dial the revolutions per minute. The latter method is more convenient if the instrument is correct, but frequent calibration and adjustment are necessary and a correction must usually be applied to values thus observed.

The revolution-counter is more accurate, and, while not so convenient, is to be preferred.

166. **Measurement of Power.**—The power of the wheel may be determined by placing a special brake pulley on the turbine shaft and applying a resistance by means of a Prony brake or some other form of dynamometer. This resistance is then measured by some form of scales (see Figs. 225, page 353, and 226, page 354 and 229, page 359). The power thus consumed by the friction of the brake can be calculated by equation (132)

$$(132)\quad P = \frac{2\pi\, l\, n\, S}{33000} \text{ in which}$$

P = Horse power.

l = length of lever or brake arm from center of revolution, in ft.

n = revolution per minute.

π = ratio of the circumference to the diameter of a circle = 3.1416.

S = weight on the scale in pounds.

This is the method applied in all laboratory work (see Fig. 225, page 353) and is that used at the Holyoke testing flume.

As the brake does not involve generator efficiencies, it is the simplest, most accurate and direct method. The Alden Absorption Dynamometer was developed and has been extensively used for this purpose.

This is a type of friction brake which usually consists of a series of smooth cast iron discs, keyed to the power shaft and revolve with it. Between these discs are stationary housings which bear upon the hubs of the discs, and each supports a pair of thin copper plates which are in contact with the cast iron discs on either side of the housing. A system of piping admits water for circulation between the unit, and the pressure of water controls the pressure of the copper plates upon the discs. A second system of piping circulates oil for lubricating the surface of copper plate and iron disc which are in contact. The housing is connected by a system of levers to a weighing machine, by means of which the power absorbed can be computed as in the case of the Prony brake.

Such a dynamometer has been used by Mr. Allen to absorb 4100 H. P., developed by a pair of turbines operating under a head of 110 feet.*

When wheels are tested in place, it is sometimes more convenient, and often essential, to determine the power output from the current generated by electrical units, which, when measured by aid of the known efficiency of the generator, will give the actual power of the wheel. If these units be direct-connected so that little or no transmission loss is involved, and if the generator is new and its efficiencies have been accurately determined, the errors involved by this method are comparatively small. The transmission of the power before measurement through gearing, through long shafts and bearings or by other means, involves losses, the uncertainties of which must be avoided if accuracy is essential.

167. Efficiency.—The efficiency of a machine is the ratio of energy delivered by the machine to that which was supplied to it and it may have various significations.

In an impulse wheel (see Section 131) the theoretical energy of the water in the forebay in foot pounds per second is:

$$E = qwh \tag{1}$$

The energy just inside the outlet of the pipe is

$$E_p = qw\,(h' + h'') \tag{133}$$

The energy of the jet is

$$E_j = \frac{qwv^2}{2g} \tag{77}$$

* See the Testing of Water Wheels after Installation, by Professor C. M. Allen; Trans. A. S. M. E. Vol. 32 p. 275.

and the theoretical power delivered to the bucket is

$$(76) \qquad E_d = \frac{qw\,(1-\phi)\,v\,(1-\cos\alpha)\,\phi\,v}{g}$$

If E_w represents the actual foot pounds of work delivered by the wheel per second then

(134) $\dfrac{E_w}{E}$ = the efficiency of the entire installation including pipe, jet, wheel, etc.

(135) $\dfrac{E_w}{E_p}$ = efficiency of the water wheel, including nozzle and buckets.

(136) $\dfrac{E_w}{E_j}$ = efficiency of the runner, and

(137) $\dfrac{E_w}{E_d}$ = hydraulic efficiency of the bucket.

In the testing of water wheels, the efficiency (135), $\dfrac{E_w}{E_p}$, is the ratio ordinarily to be determined since it involves the losses in the nozzle, jet and buckets as well as from residual energy in the water discharged by the buckets, all of which are properly chargeable to the operation of the wheel.

The efficiency represented by (137) involves only the effects of losses of energy by the water in passing over the buckets and its theoretical value is 100 per cent. for all values of ϕ. It eliminates the effect of uneconomical speed of rotation of the wheel which leaves residual lost energy in the water discharged by the buckets and not properly chargeable to bucket imperfections. It would be determined only in a detailed study or test made for the fourth purpose above mentioned.

168. Illustration of Methods and Apparatus for Testing Water Wheels.—Fig. 226, page 354, shows the apparatus used for testing turbines on a vertical shaft, by Mr. J. B. Francis to test a Swain wheel at the Boott Mills, Lowell, Massachusetts (see "Lowell Hydraulic Experiments").

The section represents a vertical turbine in the testing plant with testing apparatus in place.

The plan of the plant shows the arrangement of the Prony brake.

In these drawings P is the friction pulley; b is the brake; c are counter balances to remove the load of the brake from the wheel shaft; L is the bent lever or steel beam for transferring horizontal motion to a vertical lift; S is the scale pan for the weight; d is the dash-pot;

w is the weir for measuring the water, and r is the rack for stilling the water after leaving the wheel.

Figure 227, and Figs. 228 and 229, page 359, show the brake wheel and Prony brake details used by Mr. William O. Webber for determining the efficiency of various turbine water wheels as described by him in a paper on "The Efficiency Tests of Turbine Water Wheels." *

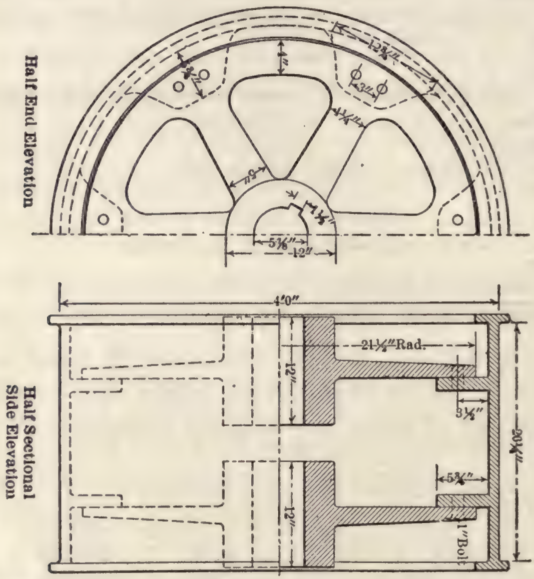

Fig. 227.—Brake Wheel Used by W. O. Webber.

169. Tests of Wheels in Place.—In April, 1903, a Leffel turbine was tested at Logan, Utah, at the station of The Telluride Power Transmission Company, by P. N. Nunn, chief engineer. The wheel was directly connected to a General Electric Company generator the efficiency of which has been determined as follows:

125 per cent. load	96.7 per cent. efficiency.
100 per cent. load	96.2 per cent. efficiency.
75 per cent. load	95.3 per cent. efficiency.
50 per cent. load	93.5 per cent. efficiency.
25 per cent. load	88.0 per cent. efficiency.

The output of this generator was used as a basis for calculating the work done by the water wheel.

* See vol. 27, No. 4, American Society of Mechanical Engineers; also Section 171, Experiments at the Holyoke Testing Flume.

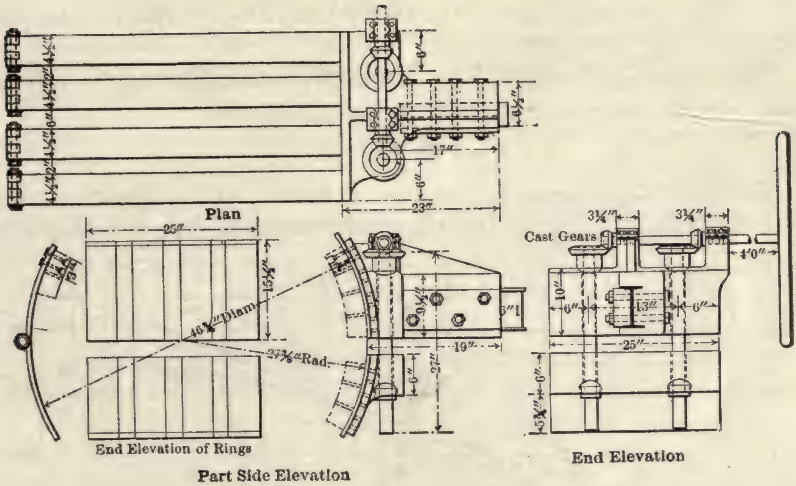

Fig. 228.

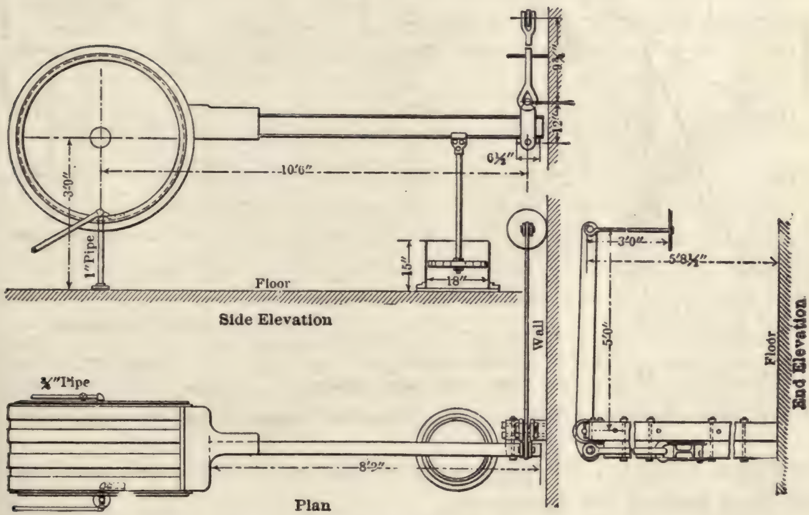

Fig. 229.—Details of Prony Brake Used by W. O. Webber (see page 358).

The results of the tests and methods of calculation are shown in Table 35, page 361, and graphically illustrated in Fig. 230, page 360.

A similar test of one of a number of wheels installed by The James Leffel Company in the plant of the Niagara Hydraulic Power and Manufacturing Company was made in December, 1903, by Mr. John L. Harper, engineer of that company. The following Table 36, page

361, is the condensed data of the test of wheel No. 8 which is also illustrated by Fig. 231, page 362.

The water was measured by a standard contracted weir 16.23 feet long and discharge computed by Francis' formula:

$$q = 3.33\ (L - 0.2h)\ h^{\frac{3}{2}}$$

The load was computed from the voltmeter and ammeter readings of two generators, Nos. 5 and 12, which were both driven by this wheel

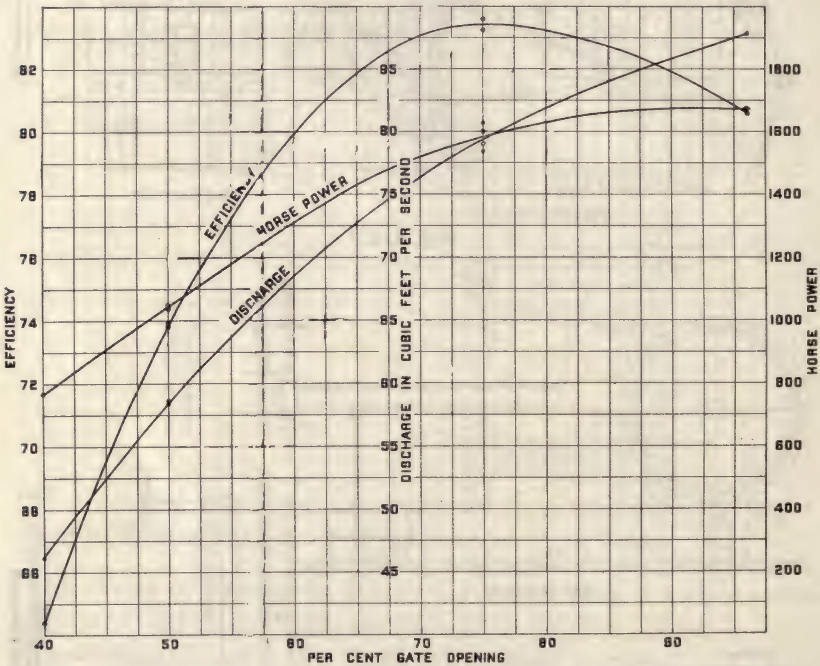

Fig. 230.—Results of Test of Leffel Wheel at Logan, Utah (see page 358).

and then corrected for the generator loss by a factor estimated from the shop tests of the generators.

The 10,500 H. P. turbine manufactured by the I. P. Morris Company for the Shawinigan Power Company was also tested in a similar manner. The graphical results of this test are shown by Fig. 232, page 363. Fig. 233, page 364, illustrates the test of a twenty-five inch Victor High Pressure Turbine, manufactured by the Platt Iron Works Company, at the Houck Falls Power Station at Ellensville, New York.

The results of various tests at the Holyoke testing flume, collected from divers sources, are given in the appendix. The later tests have

TABLE 35.

Test of High Head Leffel Horizontal Turbine at Logan Station of Telluride Power Trans. Company, Logan, Utah. Efficiency of Test at Constant Speed, April 28, 1903.

P. N. Nunn, Chief Engineer.

Gate opening	0.75	0.50	0.40	0.50	0.75	0.96
Head on 15 feet weir in feet	1.394	1.132	0.969	1.129	1.368	1.475
Discharge of weir in cubic feet per second	81.85	59.76	47.27	59.66	79.55	88.94
Leakage around weir in second feet	0.85	0.85	0.85	0.85	0.85	0.85
Exciter water in second feet	1.98	1.98	1.98	1.98	1.98	1.98
Water through turbine in second feet	80.72	58.63	46.14	58.53	78.42	87.81
Pressure at shaft center in pounds per square inch	86.5	87.3	87.5	87.2	86.5	86.2
Effective head above shaft center in feet	199.3	201.2	201.6	200.9	199.3	198.6
Vacuum head measured in feet	10.4	10.6	10.8	10.6	10.4	10.3
Total working head in feet	209.7	211.8	212.4	211.5	209.7	208.9
Theoretical horse power	1921	1409	1112	1405	1866	2082
K. W. output at switch board	1152	739	500	737	1123	1210
Generator efficiency	0.965	0.952	0.935	0.952	0.965	0.967
Brake horse power of turbine	1600	1041	717	1038	1560	1677
Efficiency of turbine	0.833	0.738	0.644	0.739	0.836	0.806
Gate opening	0.75	0.50	0.40	0.50	0.75	0.96

NOTE—Speed, 400 R. P. M. (normal).

Generator efficiency taken from test of machine made by The General Electric Company. (Record of test in office of chief engineer.)

TABLE 36.

Test of a Double Horizontal Leffel Turbine Installed in the Plant of the Niagara Hydraulic Company, Niagara Falls, N. Y.

	Gate Opening.		
	.45	.7	1.0
	Dec. 5th		Dec. 6th
Time	3:21 p. m.	5:01 p. m.	4:59 p. m.
Hook gauge reading (corrected)	1.365	1.978	2.257
Discharge of wheel by Francis' formula	84.76	146.6	178.3
Head on turbine	213.0	212.4	212.7
Hydraulic horse power	2045	3528	4320
R. P. M.	255	259	250
*Generator No. 5 **			
Volts	178	178	184
Amperes	5065	5020	5833
Efficiency	.92	.92	.92
Horse power taken from wheel by generator	1314	1302	1563
Generator No. 12 †			
Volts	Friction	12200	13000
Amperes	Load	57.7	60.5
Efficiency	Only	.95	.955
Horse power taken from wheel by generator	17	1720	1912
Total horse power output of wheel	1331	3022	3475
Efficiency of wheel	.651	.856	.805

* Generator No. 5 is a G. E. 5000 A. 715 V., D. C. machine.

† Generator No. 12 is a Bullock 1000 K. W., 3 phase A. C. generator.

been furnished by manufacturers and represent the best results of modern turbine manufacture.

170. **Desirable Extension of Holyoke Tests.**—Attention has previously been called (see Section 161, page 349) to the fact that it is very desirable to extend the tests to the limit of no revolutions and runaway

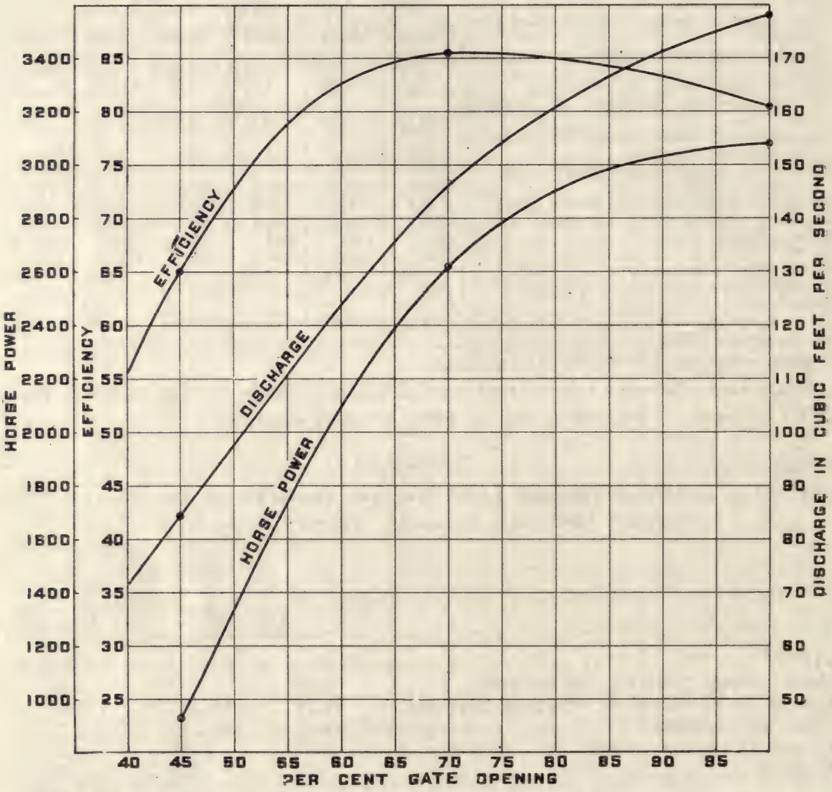

Fig. 231.—Results of Test of Leffel Wheel at Niagara (see page 359).

speed in order to check the intermediate results and give a broader basis for wheel analyses and selection.

Figure 234, page 365, shows the results of tests made on a forty-eight inch Victor turbine. These tests are rather more complete than ordinarily carried out at the Holyoke testing flume, yet the range of the actual tests compared with the possible range of such tests is comparatively small. In this particular case tests at runaway speed were made at three gate openings but no tests were available under stationary conditions. The advantages of a test at the extreme limits

are perhaps best shown by reference to Fig. 235, page 366.* In this particular case only one test was practicable near the points of maximum efficiency and power, and this test was at the speed for which the unit was designed and practically correspond to a speed of fifty R. P. M. under one foot head. However, the determination of the torque at fixed gate openings and under stationary conditions and at runaway speed (with no torque) permitted a completion of the experimental curves in a manner which although approximate gave a

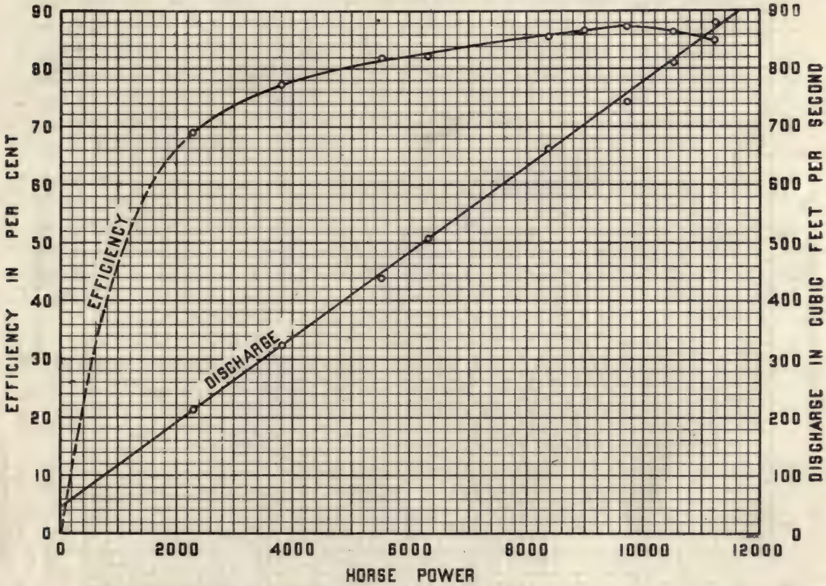

Fig. 232.—Results of Tests of 10,500 H. P. Turbine at Shawinigan Falls, Canada (see page 360).

very fair basis for determining with reasonable accuracy the probable operation of the wheel under all conditions of speed.

171. **Holyoke Tests and Actual Results in Place.**—The cost of installation of large plants is so great that it is necessary to determine the power and efficiency that can be expected within a comparatively small margin before the plant is constructed. In consequence, the testing of actual wheels with their permanent chute cases, gates and guides, whenever they are sufficiently small for that purpose, or of homologous wheels similarly equipped, whenever the turbines are too

* See The Investigation of the Performance of a Reaction Wheel, by Prof. R. L. Daugherty. Proc. A. S. C. E., Vol. XL, No. 8, page 2477.

large for such purposes, is essential. The Holyoke testing flume is suitable for the satisfactory testing of wheels of perhaps forty inches or even larger for wheels of small capacity. For larger wheels, the results at Holyoke are not fully satisfactory on account of the fact

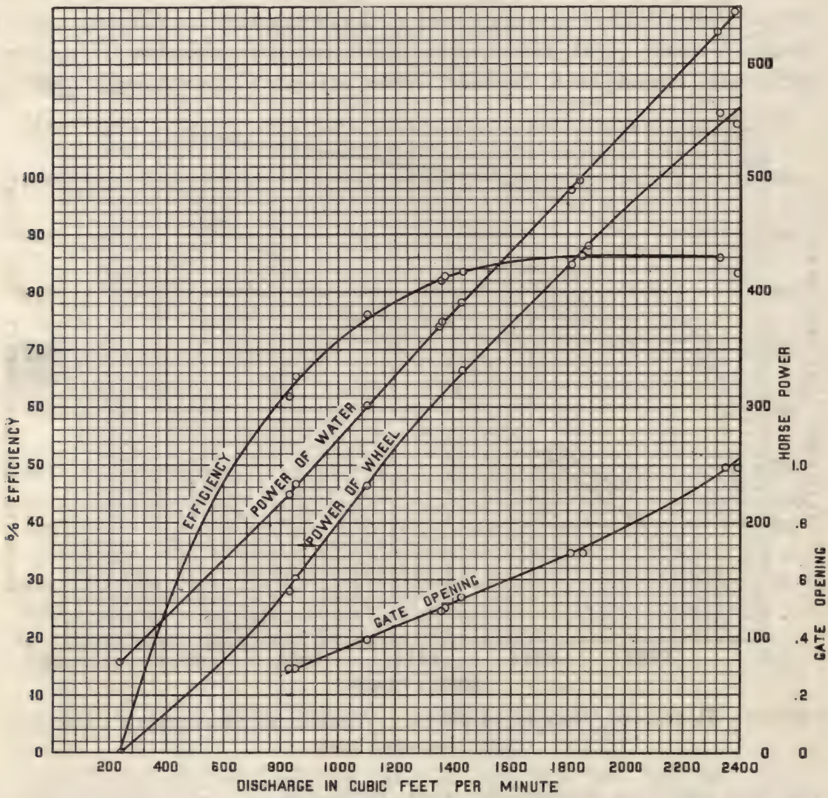

Fig. 233.—Test of Victor Turbine at Houck Falls, New York (see page 360).

that the quantity of water is limited by the size of the supply pipe, the head is reduced by the large amount of water needed, and the discharge is throttled by the arrangement of the flume. In all such cases, therefore, the best results will be attained through the tests of homologous wheels.

In many cases homologous wheels are tested at Holyoke in vertical cases, different in design from those which will be used, especially when the wheels are to be set with a horizontal shaft, either single, twin, quadruplex, etc.

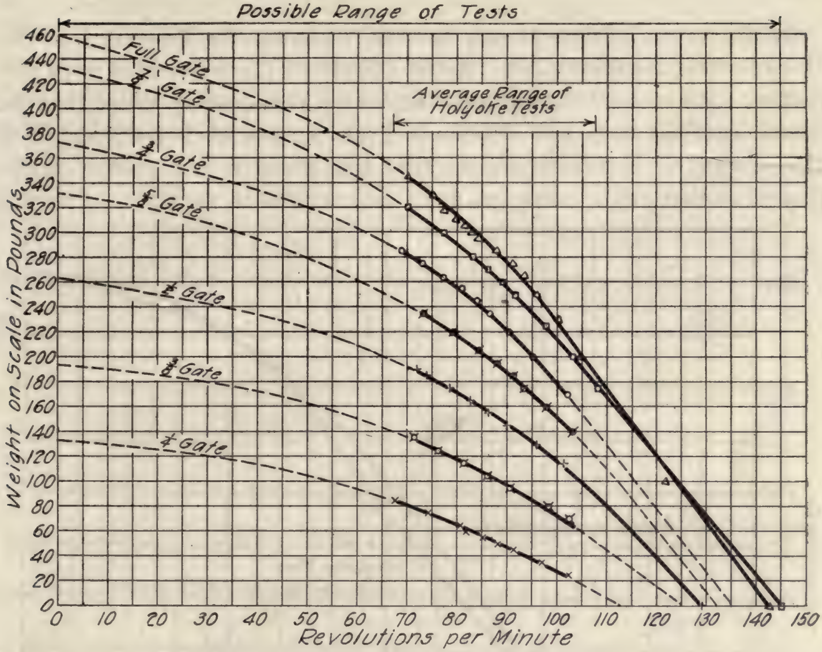

Fig. 234.—Speed-Resistance Relations for a 48-inch Victor Turbine Under 13-foot Head (see page 363).

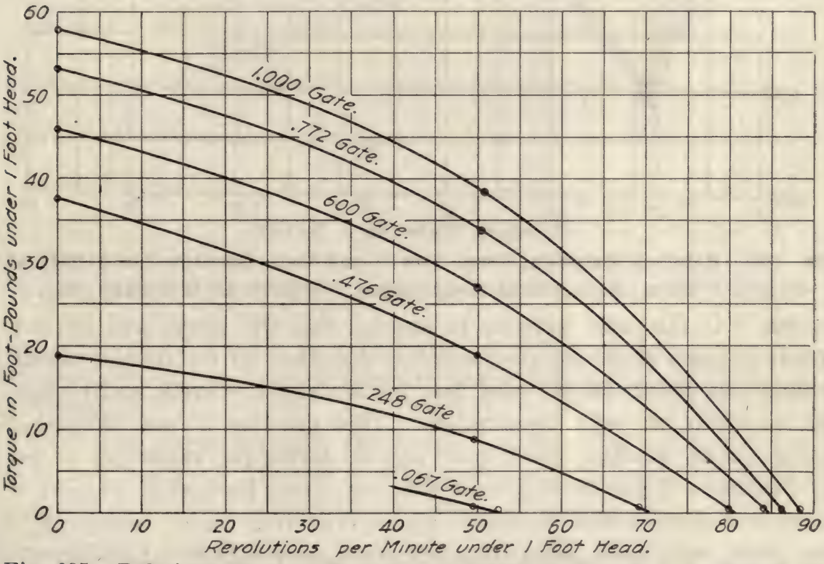

Fig. 235.—Relations Between Torque and Speed of 27-inch I. P. Morris Reaction Turbine (see page 363).

In the actual design or selection of turbines, the relation between Holyoke test results and the actual results in place becomes very important for to secure the highest efficiency it is necessary to see that the wheel installed does not develop a great excess of power, and it is almost equally important to see that the wheel is not greatly under power, as that might often involve a serious loss in capacity. For this

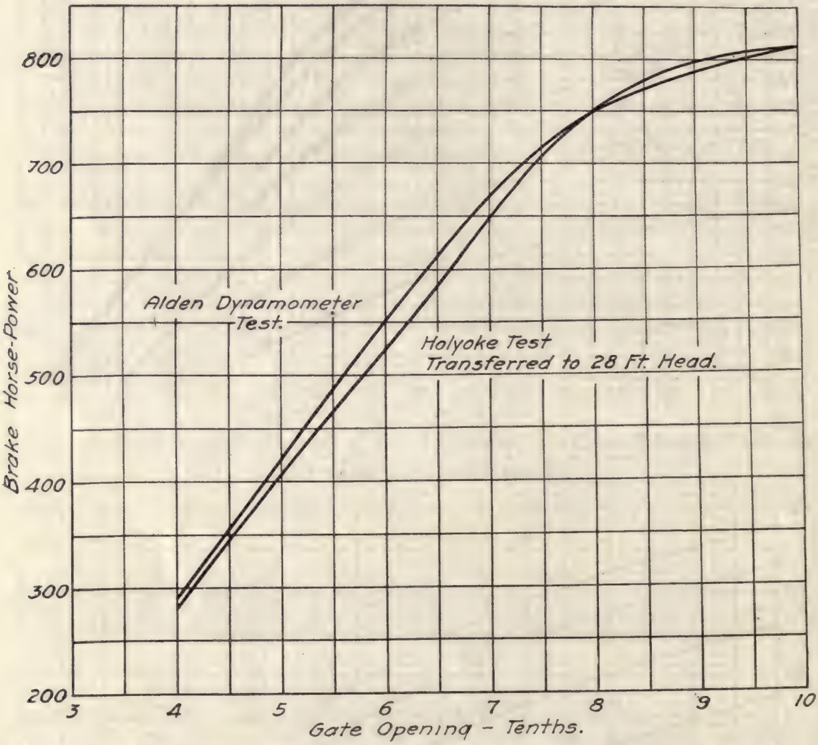

Fig. 236.—Relation Between Horse Power and Gate Opening for a Pair of 36-inch Wheels Under 28-foot Head and at 200 R. P. M. (see page 363).

reason it is the usual practice to assume that the wheel will develop power in place about five per cent. less than the test for simple vertical wheels and about ten per cent. less for horizontal wheels, and to base the selection on such assumption. This practice is not altogether borne out by results. In Figure 236 is shown the results of a test by Professor Allen on a pair of thirty-six inch horizontal wheels set three and three-tenths diameters apart, operating under twenty-eight foot head, with draft tubes changing from a circular to an oval cross-section and with constantly increasing area. This comparison of re-

sults show an agreement at full gate and comparatively small differences at part gate.* Figure 237 shows the results of Professor Allen's test of a pair of forty-eight inch horizontal wheels set 4.25 diameters apart and under a forty foot head.* The apparent increase

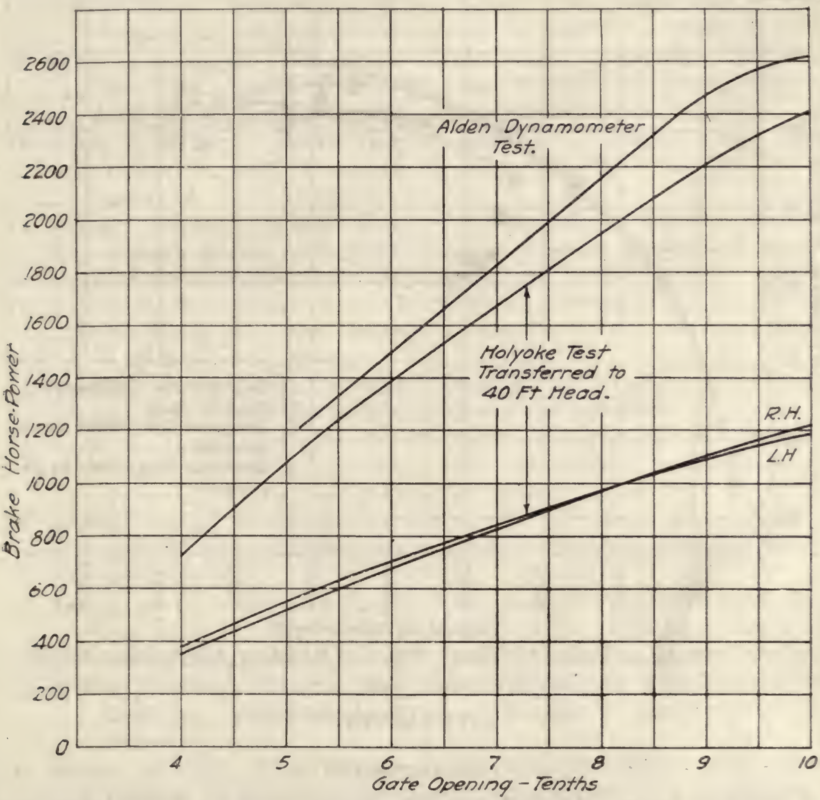

Fig. 237.—Relation Between Horse Power and Gate Opening for a Pair of 48-inch Wheels Under 40-foot Head and at 200 R. P. M.

in power in these wheels is believed to be due to the fact that for wheels of this size and capacity the Holyoke tests are apt to give too low results.

Figure 238, page 368, is a test made by Mr. C. T. Main, of the wheels installed by S. Morgan Smith Company at Rainbow, Montana. The diagram shows a comparison of efficiency power curves between

* The Testing of Water Wheels After Installation by Prof. C. M. Allen. Trans. A. S. M. E., vol. 32, p. 275.

the estimate from the Holyoke test, the results guaranteed, and the actual results obtained. These units, six in number, have double runners and cast iron scroll cases and operate under 107 foot head.

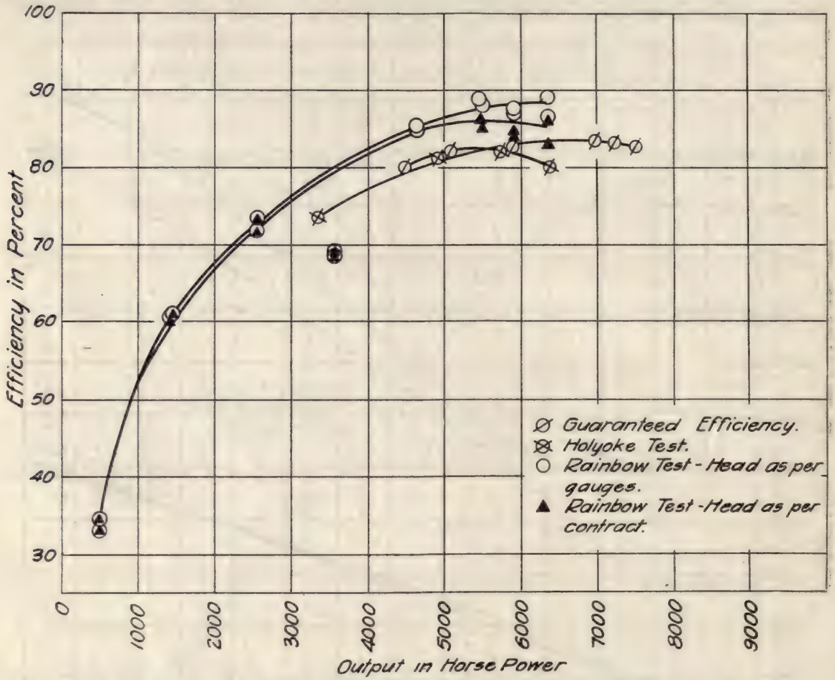

Fig. 238.—Efficiency Tests of a Water Wheel at Rainbow, Mont. (see page 367).

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CHAPTER XIII

TURBINE ANALYSIS AND SELECTION

172. **Symbols Used in This Chapter.**—The letters and symbols used in this chapter have the following significance:

D = Diameter of wheel in inches.
e = Efficiency.
h, h_h, h_1 = Effective head under which the wheel is to operate.
n = R. P. M. = Revolutions per minute.
n_h, n_{16}, n_1 = R. P. M. of homologous wheels of equal diameters under heads of h, sixteen and one feet respectively.
P = Horse power of wheel at any given head.
P_h, P_{16}, P_1 = Horse power of homologous wheels of equal diameter under heads of h, sixteen and one feet respectively.
q = Discharge of wheel under given head in cubic feet per second.
q_h, q_{16}, q_1 = Discharge of homologous wheels of equal diameter under heads of h, sixteen and one feet respectively.
$\triangle$ = Speed coefficient = R. P. M. of one inch wheel under one foot head.
K = Discharge coefficient = Discharge of one inch wheel under one foot head in cubic feet per second.
p = Power coefficient = Horse power of one inch wheel under one foot head.
$\mathfrak{P}$ = Specific power = Horse power of a wheel at one revolution per minute under one foot head.

173. **Reduction of Data to Uniform Head.**—An examination of the results of any turbine test will show that under the ordinary conditions of conducting such a test the head will vary during the test to a greater or less extent, and will commonly decrease as the gate opening increases and more water is discharged by the turbine. It is evident, therefore, that in order to determine how the wheel will actually operate under a uniform head, it will be necessary to reduce the experimental data taken under varying heads from the test results to the equivalent results which would be secured under the head at which the wheel is to operate. If, for example, the head h_h is the desired operating head, the various factors can be reduced, according to the principles discussed in Chapter XI by the following formulas:

For example, take Experiment No. 80 of Table 80 (Appendix), in

which the following data are given, which it is desired to reduce to a head of sixteen feet:

Number of experiment.	Gate opening, inches.	Proportional discharge (discharge at full gate with highest efficiency=1).	Head acting, feet.	Duration of test in minutes.	Revolutions per minute.	Discharge in cubic feet per second.	Horse-power developed.	Percentage of efficiency.
1	2	3	4	5	6	7	8	9
80	2.75	1.010	17.49	4	130.0	64.45	91.97	71.94

The data, formulas and computations will be as follows:

Experimental Data	Formulas	Calculations	Results
$h = 17.49$			$h_h = 16$
$n = 130$	(93) $n_h = \frac{n\sqrt{h_h}}{\sqrt{h}}$	$n_{16} = \frac{130\sqrt{16}}{\sqrt{17.49}}$	$n_{16} = 124.3$
$q = 64.45$	(97) $q_h = \frac{q\sqrt{h_h}}{\sqrt{h}}$	$q_{16} = \frac{64.45\sqrt{16}}{\sqrt{17.49}}$	$q_{16} = 61.61$
$P = 91.97$	(108) $P_h = \frac{P\, h_h^{\frac{3}{2}}}{h^{\frac{3}{2}}}$	$P_{16} = \frac{91.97 \times 16^{\frac{3}{2}}}{17.49^{\frac{3}{2}}}$	$P_{16} = 80.4$
$e = 71.94$			$e = 71.94$

As the speed ratio has been kept constant in these calculations, the efficiency will remain constant for this and all similar reductions. Each experiment in the test shown in Table 80 has been reduced in the above manner and has been platted in three sets of curves (see Fig. 239, page 374) by which the common relations of the efficiency, power and discharge of this wheel at various speeds for various gates and under a sixteen foot head are shown. From this diagram it will be seen that under a sixteen foot head, the most efficient speed of operation will be at about 173 R. P. M., and that either an increase or a decrease in speed will result in a decrease of the maximum efficiency. The relations of efficiency to power at this efficient speed and at speeds of 160 and 185 R. P. M. as derived from this diagram are shown in Fig. 240. page 376.

Diagrams showing in detail the results to be anticipated from the use of this wheel under any other head can be made in a similar manner; or this diagram can be used directly for the calculations of such data by applying the formulas used for each point to be considered.

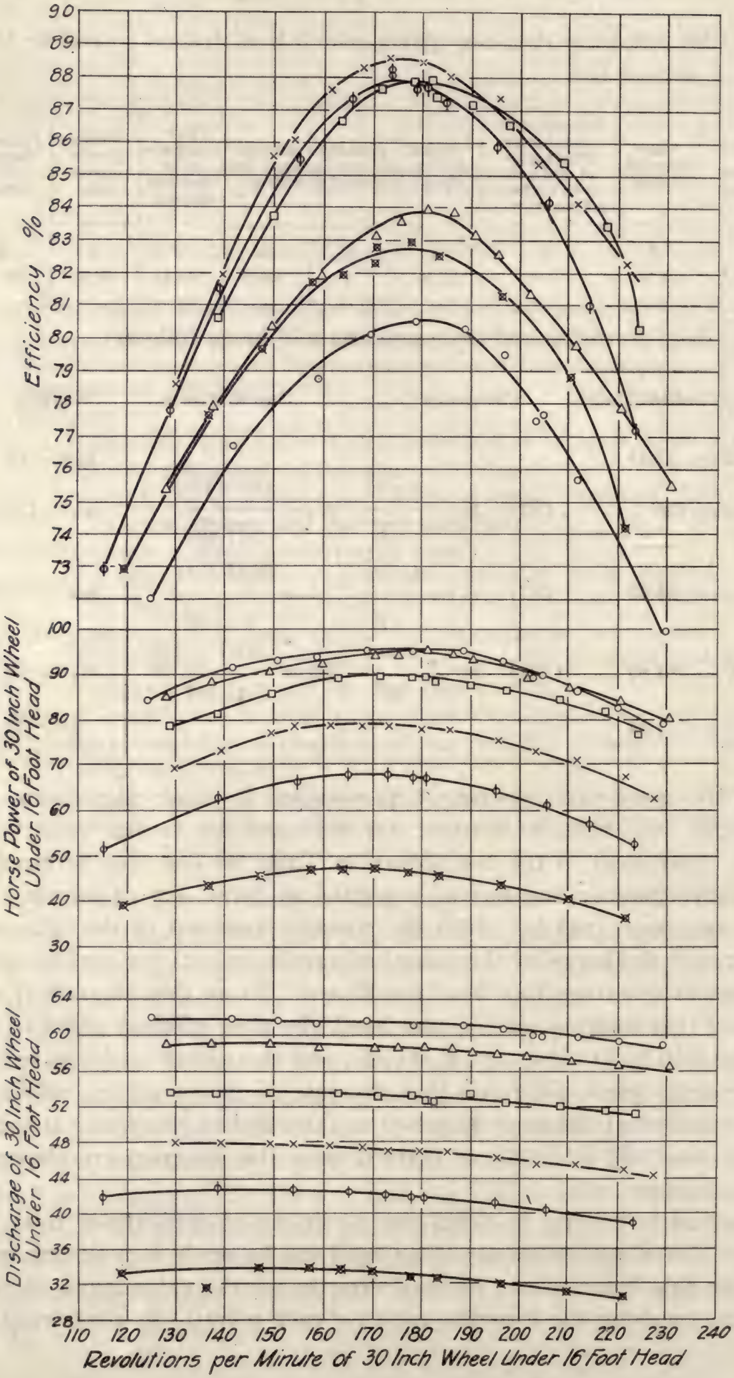

Fig. 239.—Diagram of Results of Test of 30-inch Wheel After Reduction to Basis of 16-foot Head (see page 373).

174. Reduction of Data to One Foot Head.—As the variations in speed n, discharge q, and power P are in proportion to $\sqrt{h}$, $\sqrt{h}$, and $h^{\frac{3}{2}}$, respectively, and as these functions of h become unity for a head of one foot, it is frequently desirable to select a head of one foot ($h = 1$) as a basis for graphical turbine analyses. This renders such a diagram more readily available for general use. When this is desired, the formulas used and the calculations as applied to the data considered in Section 173 are as follows:

Data	Formulas	Calculations	Results
$h = 17.49$			$h_h = 1$
$n = 130$	(94) $n_1 = \dfrac{n}{\sqrt{h}}$	$n_1 = \dfrac{130}{\sqrt{17.49}}$	$n_1 = 31.1$
$q = 64.45$	(99) $q_1 = \dfrac{q}{\sqrt{h}}$	$q_1 = \dfrac{64.45}{\sqrt{17.49}}$	$q_1 = 15.41$
$P = 91.97$	(110) $P_1 = \dfrac{P}{h^{\frac{3}{2}}}$	$P_1 = \dfrac{91.97}{17.49^{\frac{3}{2}}}$	$P_1 = 1.257$
$e = 71.94$			$e = 71.94$

In this, as in all cases, the efficiency e remains constant as long as the speed ratio is constant.

Figure 241, page 377, is a graphical diagram in which each test in Table 80 has been reduced to the basis of one foot head in the above manner, and platted as in the case discussed in Section 173.

This diagram can be conveniently used for the consideration of this wheel for all heads. In this case, the most efficient speed at one foot head is 43.25 R. P. M. As the efficiencies at any given speed ratio remain constant and as the speed and discharge at any other head vary as $\sqrt{h}$, and as the power at any other head varies as $h^{\frac{3}{2}}$, the efficiency, the discharge and the power at each gate opening will be equal to the diagramatic readings at the points where the efficient speed line of 43.25 R. P. M. crosses the efficiency, power and discharge lines for each gate, multiplied by the appropriate factor. That is, this diagram can be used for the determination for any head and for each gate, of the speed n, by multiplying the speed shown under the given speed conditions on the diagram by $\sqrt{h}$. The value of discharge q, can be determined by multiplying the diagramatic value of q_1 by $\sqrt{h}$, and the

value of the power P, can be determined by multiplying the value of P_1 as given on the diagram for one foot head, by $h^{\frac{3}{2}}$. For example, for each of the heads given below, the diagramatic values must be multiplied by the ratio given under each head for speed, power and discharge.

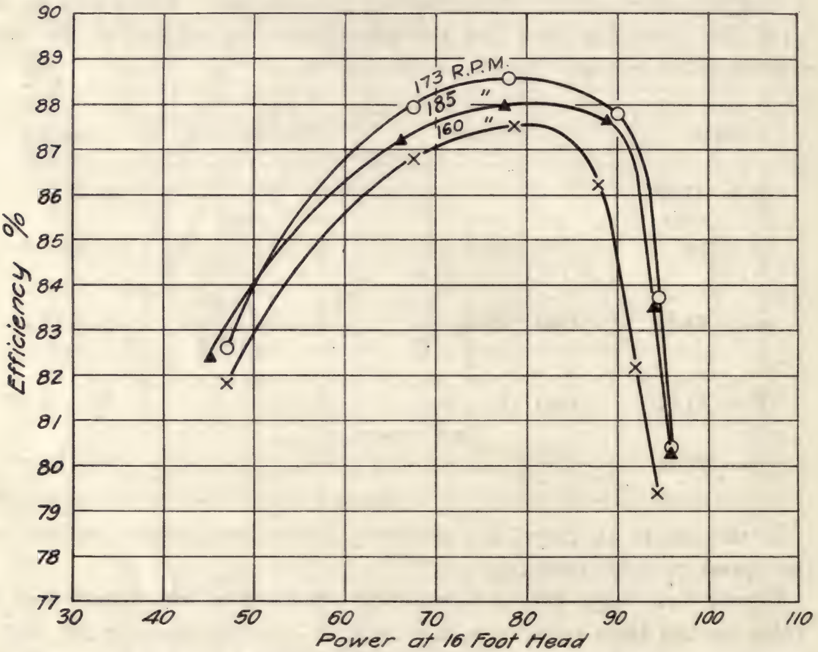

Fig. 240.—Relations of Efficiency and Power of 30-inch Wheel at Different Speeds Under 16-foot Head (see page 373).

For head of	1′	4′	9′	16′	25′	36′
Ratio of R. P. M.	1	2	3	4	5	6
Ratio of power	1	8	27	64	125	216
Ratio of discharge	1	2	3	4	5	6

Efficiency constant for all heads.

To illustrate the use of this method, the efficiency power curve for this wheel under sixteen foot, twenty-five foot and thirty-six foot head has been calculated and platted in Fig. 242, page 378.

175. **Reduction of Data to Unit Diameter and Head.**—The methods described in the last two sections are most satisfactory when considering the results which can be obtained under a certain head or under various heads from a wheel of the same characteristics and diam-

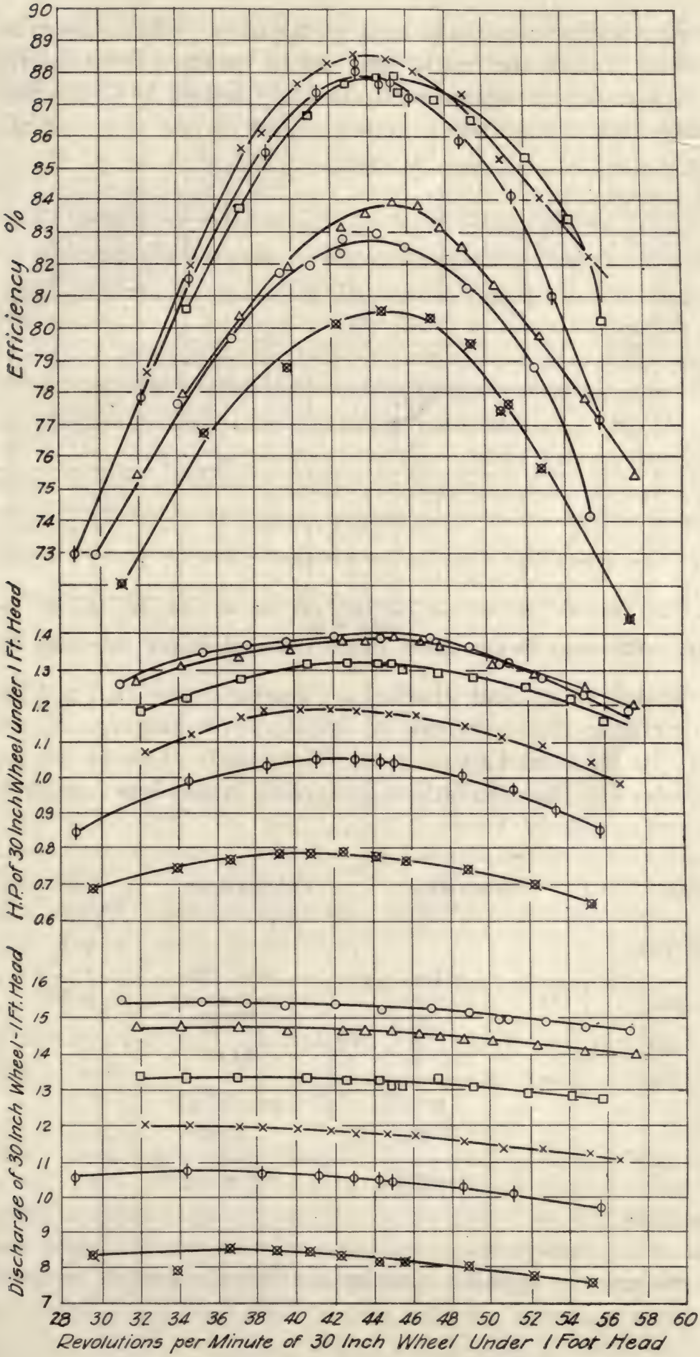

Fig. 241.—Diagram of Test Results After Reduction to Conditions at 1-foot Head (see page 375).

eter as the wheel on which the tests were made. When, however, the test is made to show the results that can be obtained from a series of wheels of homologous design with the wheel tested, it is preferable to reduce the data not only to a uniform head of one foot, but also to

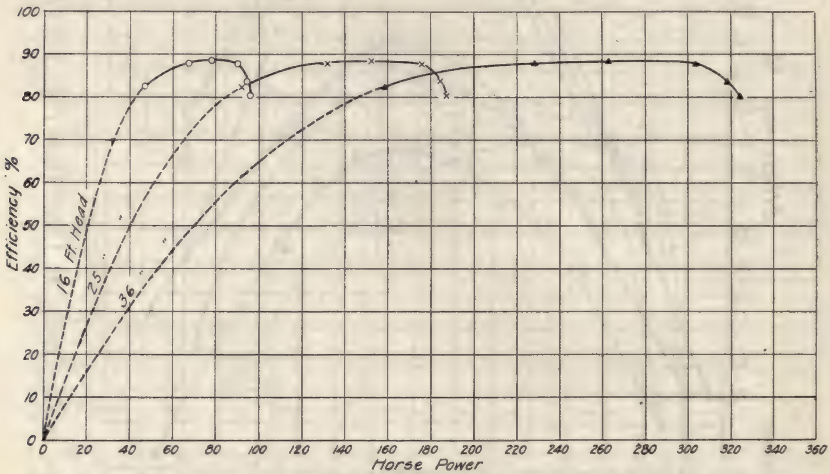

Fig. 242.—Efficiency Power Curve Under Different Heads (see page 376).

reduce the data to an ideal wheel of a diameter of one inch and by this means determine the coefficients of speed $\triangle$, of discharge K, and of power p, for all conditions of gate. This can be done by the use of the formulas and the calculations as applied to the data considered in Section 173 as follows:

Data	Formulas	Calculations	Results
$h = 17.49$			$h_1 = 1$
$n = 130$	(91) $\triangle = \dfrac{Dn}{\sqrt{h}}$	$\triangle = \dfrac{30 \times 130}{\sqrt{17.49}}$	$\triangle = .932$
$q = 64.45$	(101) $K = \dfrac{q}{D^2 \sqrt{h}}$	$K = \dfrac{64.45}{30^2 \sqrt{17.49}}$	$K = .01711$
$P = 91.97$	(113) $p = \dfrac{P}{D^2 h^{\frac{3}{2}}}$	$p = \dfrac{91.97}{30^2 \times 17.49^{\frac{3}{2}}}$	$p = .001397$
$e = 71.94$			$e = 71.94$

The efficiency e, remains constant for speed ratios as in previous cases.

Figure 243, page 380, is a graphical diagram in which each test in Table 80 has been reduced to the conditions of a one inch wheel under one foot head and platted as in the cases previously discussed.

Having reduced the test data by one of the three methods described in this and the previous two sections, the data may be platted and the points of efficiency, discharge and power for the varying speeds may be connected by average lines which will indicate the variation of the efficiency, discharge and power with the variations in speed. The results are exactly similar in each of the three cases except that there must be a change in scale. The results of the three methods of platting, based on the turbine tests given in Table 80 are embodied in Fig. 244, page 381, where the use of the three scales (when needed) gives a direct comparison of the three methods above described.

In using the unity diagram first described in this section, the formulas previously given in this section must be rearranged as follows:

$$\text{(91)} \qquad n = \frac{\triangle \sqrt{h}}{D}$$

$$\text{(91)} \qquad D = \frac{\triangle \sqrt{h}}{n}$$

$$\text{(101)} \qquad q = KD^2 \sqrt{h}$$

$$\text{(113)} \qquad D = \sqrt{\frac{P}{ph^{\frac{3}{2}}}}$$

$$\text{(113)} \qquad P = pD^2h^{\frac{3}{2}}$$

For example, if it is desired to use a wheel of this type to develop 700 H. P. under a head of sixty-four feet, then from equation

$$\text{(113)} \qquad D = \sqrt{\frac{700}{.00155 \times 512}} = 29.7 \text{ or practically 30 inches}$$

The unit speed is $\triangle = 1300$; hence for a sixty-four foot head, from equation

$$\text{(91)} \qquad n = \frac{1300 \sqrt{64}}{30} = 346.6 \text{ R. P. M.}$$

The efficiency for each gate can be determined by the efficiency at the point of intersection of the $\triangle = 1300$ line with the various gate efficiency lines. The power for each gate can be determined by the use of equation (113), taking the value of p for each gate at the intersection of the power lines for that gate with the line $\triangle = 1300$.

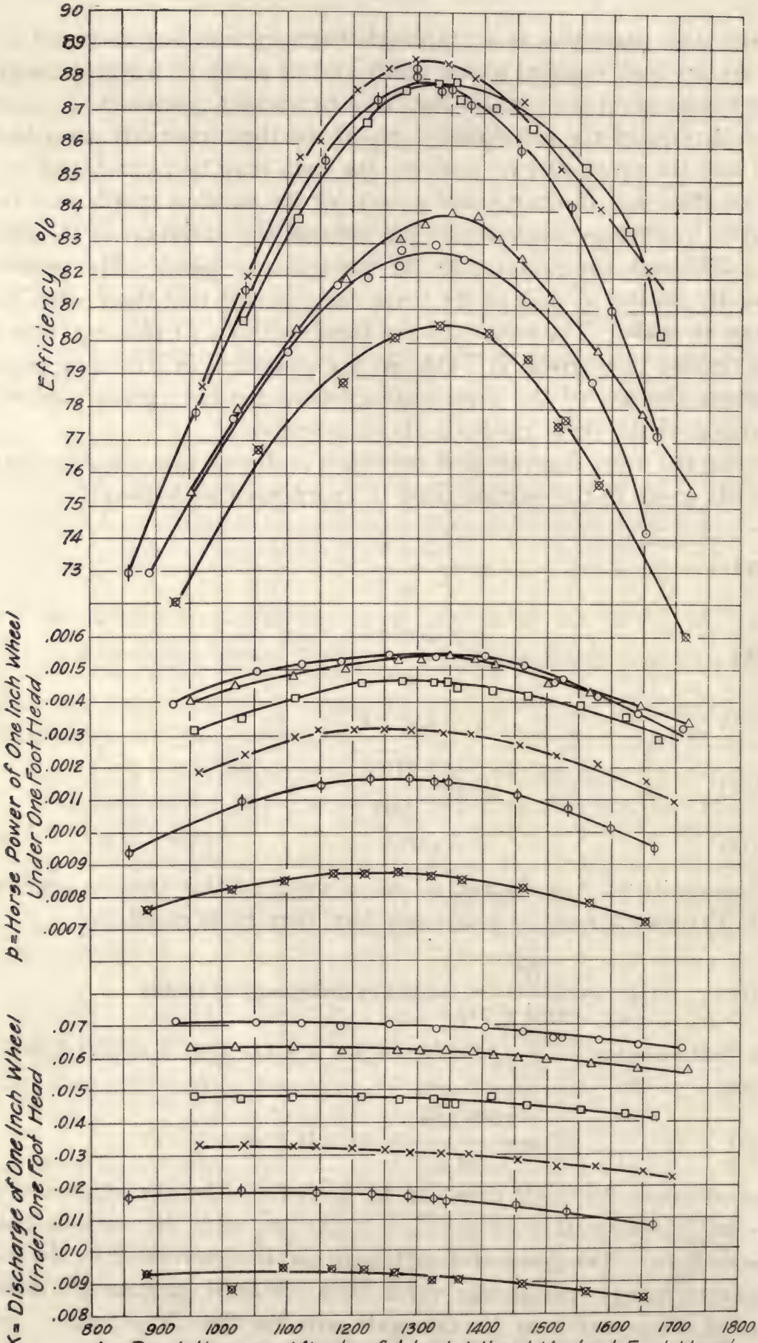

Fig. 243.—Diagram Showing Results of the Test of the 30-inch Wheel After Reduction to Conditions of a 1-inch Wheel Under 1-foot Head (see page 378).

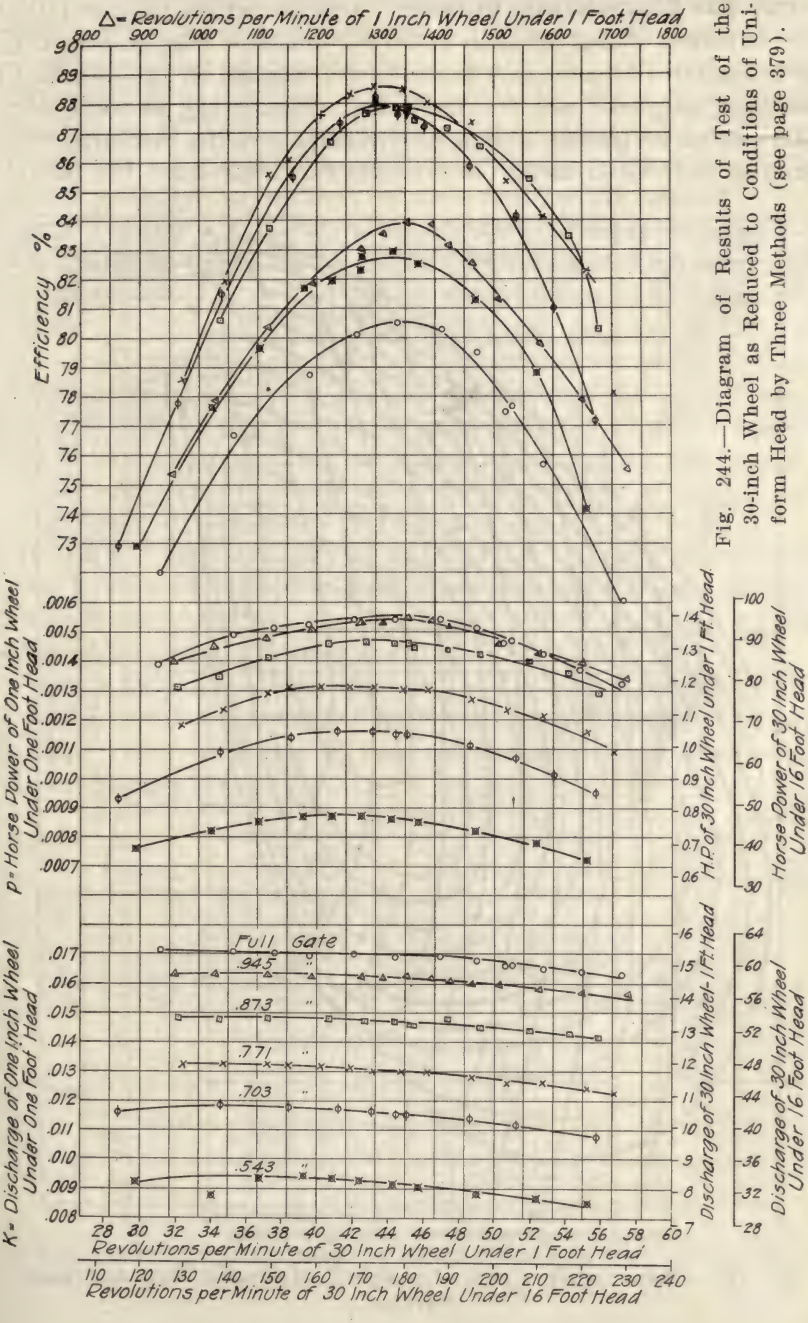

Fig. 244.—Diagram of Results of Test of the 30-inch Wheel as Reduced to Conditions of Uniform Head by Three Methods (see page 379).

176. The Characteristic Curve.—A second method of a graphical representation of the variation in turbine relations may be shown by the characteristic curve in which a single diagram is used instead of the three diagrams used in Figure 244. Figure 245 shows a characteristic curve platted with ϕ as ordinates and the discharge of a fifty-two inch wheel under one foot head as abscissae. The interre-

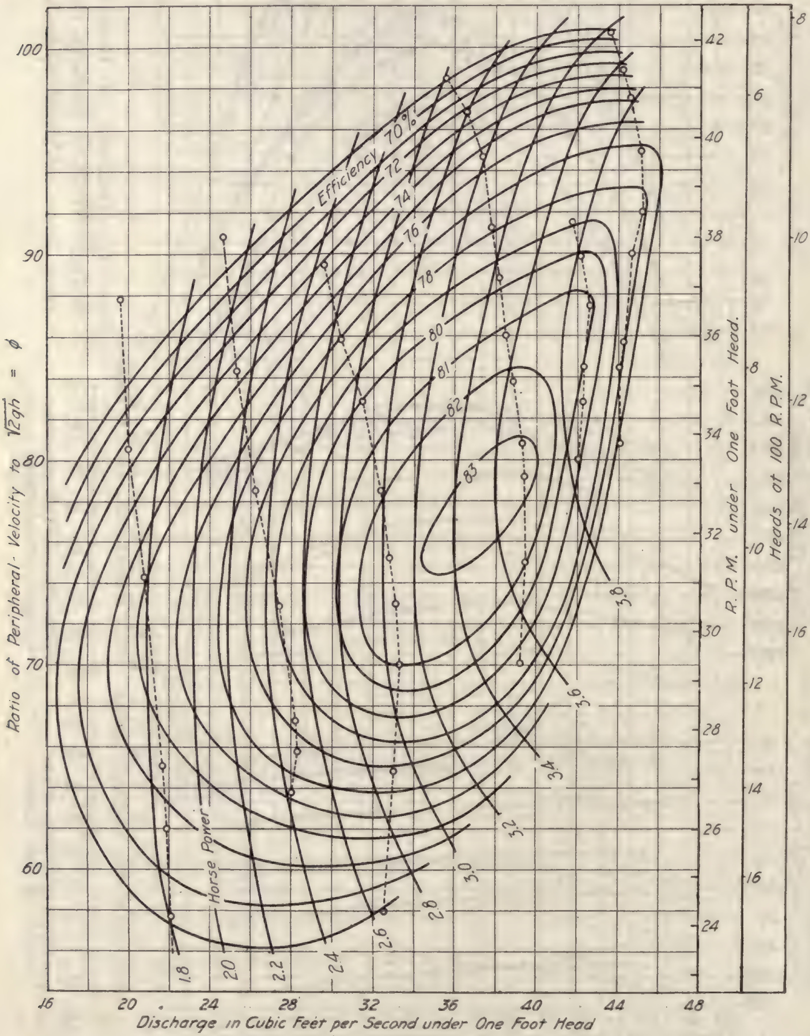

Fig. 245.—Characteristic Curve of a 44-inch "Improved New American" Turbine.

lations between the various characteristics of speed, discharge and power are perhaps best shown by the characteristic curve, but the form of analytical diagram shown in Figs. 239, 241, 243 and 244 has special advantages and is commonly used for turbine analyses. In most cases, the variation in discharge under variations of speed and gate are not of sufficient importance to be noted, and therefore in most working diagrams the sets of curves showing these relations are omitted.

177. **Variations of Relative Speed With Changes in Head.**—In many places under low head conditions, the maximum head occurs with low water flow. As the flow of the stream increases it is frequently necessary to maintain or perhaps reduce the elevation of the head water, while the tail water will rise with the increase in the river discharge. Under such conditions, with an increased flow a decrease in head will result. With such variation in head it is important to select such an operating speed for the wheels to be installed that they will work as efficiently as possible under the entire range of heads. This is commonly accomplished by the selection of a speed n, which will secure the maximum efficiency at an intermediate head. For example at the Prairie du Sac plant of the Wisconsin River Power Company, each unit consists of four sixty-four inch Allis-Chalmers wheels, and will operate under a constant speed of 107 R. P. M.

At high water an extreme minimum head of twenty feet may obtain, while the maximum head at low water may reach an extreme of thirty-four feet. The maximum efficiency of these wheels is secured with a speed coefficient

$$\triangle = \frac{Dn}{\sqrt{h}} = \frac{64 \times 107}{\sqrt{28}} = 1295$$

For other heads the values of $\triangle$ will be as follows:

Head (h)	Speed Coefficient ($\triangle$)
34	1175
32	1210
30	1250
28	1295
26	1345
24	1400
22	1460
20	1535

Figure 246, page 384, shows the ordinary form of analytical diagram used in current practice, from which (as previously noted) the dis-

charge curves are omitted. On this diagram are platted vertical lines locating the relative speed coefficients for the various heads from twenty to thirty-four feet as shown in the above table. The points at which these lines cross the efficiency curves at various gates will give

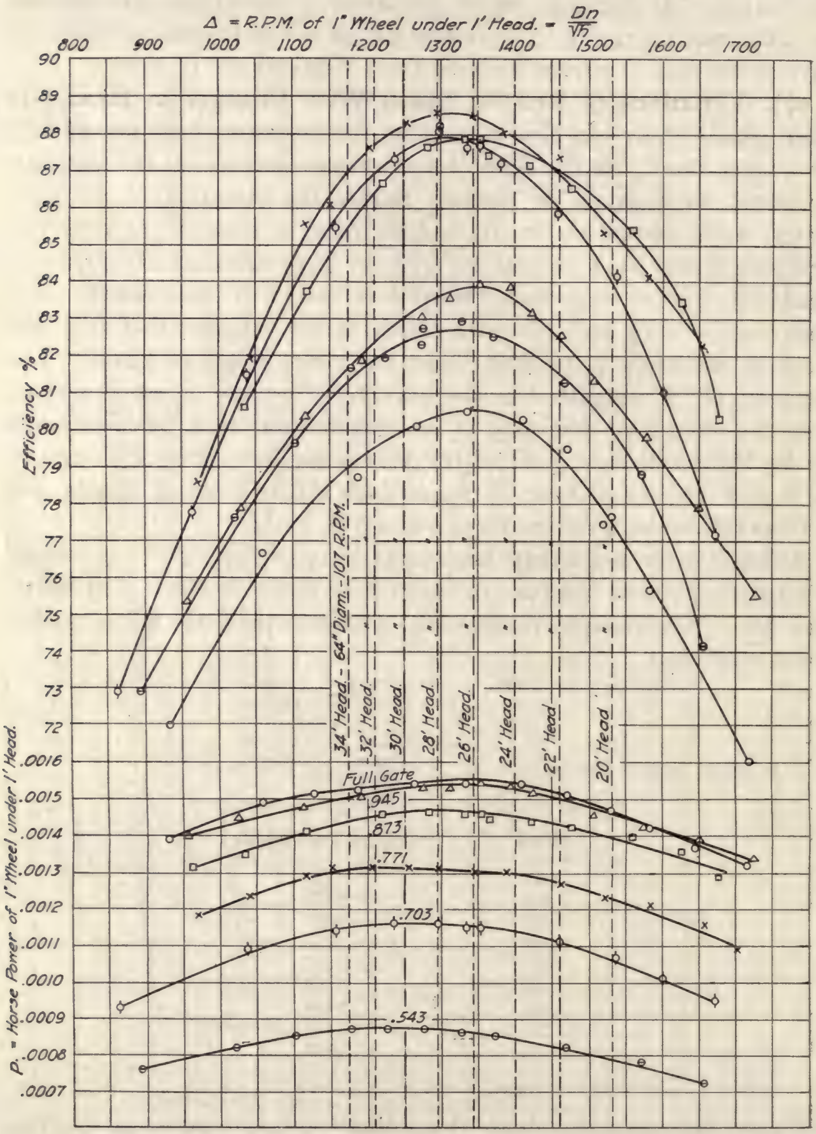

Fig. 246.—Ordinary Form of Diagram of Turbine Analysis (see page 383).

directly the efficiency at that gate and the relative speed. The coefficient of power which will obtain at the relative speed is indicated by the intersection of these same lines with the curves of the graphical diagram indicating the power coefficients at various gates, and the power for each point can be determined by the formula

$$P = pD^2h^{\frac{3}{2}} \tag{113}$$

Figure 247 shows the resulting relations of power and efficiency of a single runner under the various heads above considered.

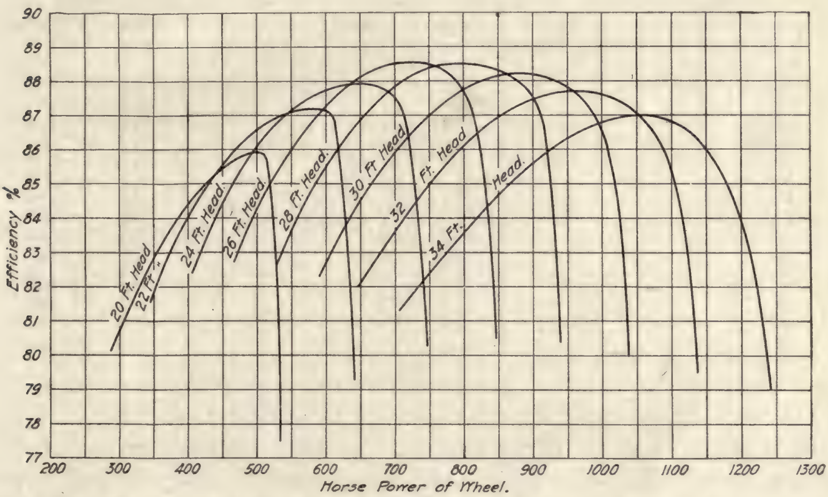

Fig. 247.—Relations of Power and Efficiency of a Single Runner Under Various Heads.

178. **Specific Power Relations.**—The forms of diagrams described in the last four sections make clear the relations of speed n, discharge q, power P, and the speed coefficient $\triangle$, discharge coefficient K, and power coefficient p, for any homologous series of wheels.

In the analysis of turbines, it is also essential to show the interrelations of one series of turbines with other series having different characteristics. For this purpose, the use of the specific power coefficient $\mathfrak{P}$ becomes desirable.

The relations of specific power to efficiency are shown in very satisfactory manner in the diagram, Fig. 218, page 330, which is reproduced as Fig. 248, page 387. In this figure are shown the relations of efficiency to specific power for each of the modern wheels for which tests are given in the appendix, Tables 72 to 101 inclusive. Each Roman numeral (I, II, etc.) represents the results of the tests of the

wheel of a single manufacturer, with the exception of diagram IV, where three manufacturers are represented. The practicing engineer should make a similar diagram on a much larger scale, and should plat thereon the tests of the wheels which may be submitted from time to time for his consideration.

The Holyoke tests of the wheels for which the efficiency-specific power curves are shown in Fig. 248 can be found in the tables of tests as given in the following tabular index.

Wheel	Table No.	Page	Wheel	Table No.	Page
I A	93	798	IV A	77	767
I B	101	812	IV B	86	784
II A	73	759	IV C	88	788
II B	78	768	IV D	85	782
II C	82	776	IV E	87	786
II D	91	794	V A	72	757
II E	100	810	V B	80	772
III A	74	761	V C	89	790
III B	76	765	V D	94	799
III C	79	770	V E	99	808
III D	81	774	VI A	75	763
III E	83	778	VI B	84	780
III F	97	805	VI C	96	803

In view of the previous discussion of this subject, the use of Fig. 248 is quite obvious. If for example, we desire to develop 100 H. P. at 200 revolutions per minute under a sixteen foot head, we will have the conditions:

$$\mathfrak{P} = \frac{n^2 P}{h^{\frac{5}{2}}} = \frac{200^2 \times 100}{1024} = 3900$$

From Fig. 248 we find that from the diagrams the following wheels will give an efficiency above eighty-eight per cent. with this specific power: namely, II C, III D, III E and VI B. If a graphical analysis of each of these wheels be made on the basis of Sections 174 or 175 and compared, a diagram similar to Fig. 249, page 388, can be made which shows that under the conditions named the wheel III E seems best suited to fit the conditions of the problem.

It should be noted that in the tables for modern turbines, Tables 72 to 101 inclusive, the tests are given essentially in the inverse order of the magnitude of $\mathfrak{P}_e$, that is, the wheels of low specific power or low capacity are first in order.

179. Turbine Coefficients for Most Efficient Speed of Operation.— As a basis for preliminary selection, that is, for determining in general

what type of turbine will probably be best suited for a particular condition, the sub *e* values of the coefficients for various types of wheels are very useful. In Table 37, page 389, are shown such values for standard wheels of various manufacturers. These values are the average of values determined from the catalog speed, power and discharge for a considerable range of head and diameter. In Table 37 are also shown such values calculated from the actual tests of individual wheels.

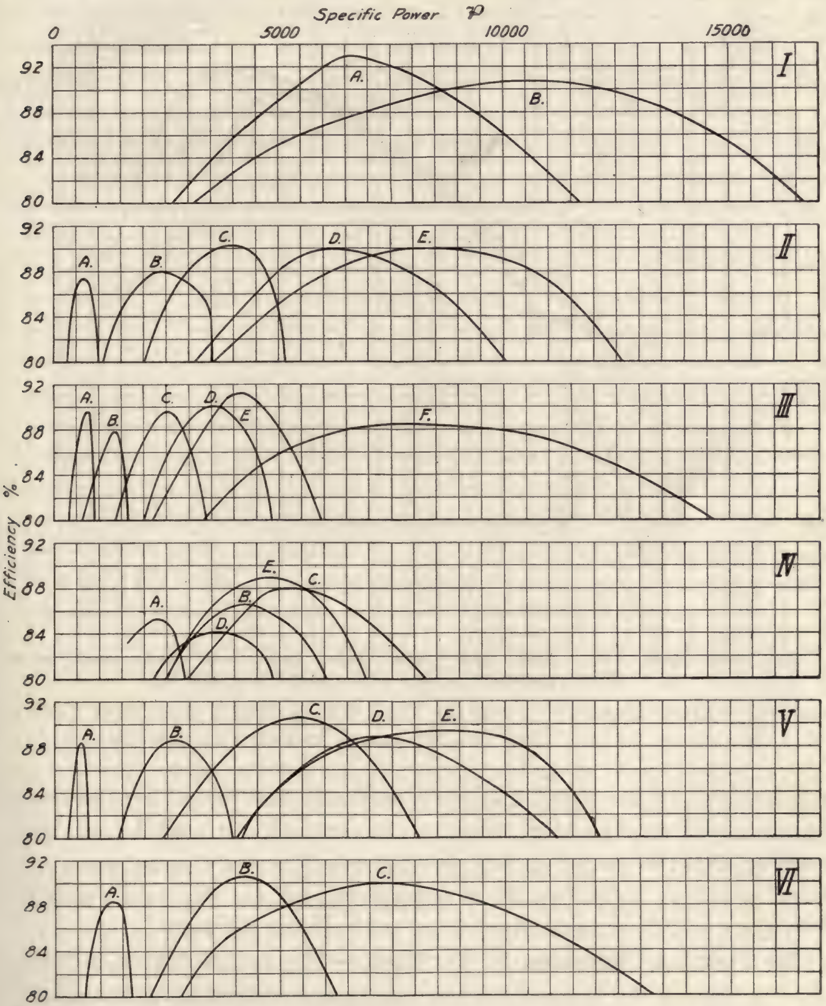

Fig. 248.—Relations Between Efficiency and Specific Power for Various Turbines (see page —).

It is evident from the various diagrams previously discussed, that a departure from these values of the coefficients may mean a decrease in efficiency. However with small variations in speed this decrease is slight, and frequently has to be made in order that a wheel may be secured which will operate under certain desired conditions. This table, therefore, is indicative of the wheel or wheels which will best suit the various conditions. From such a table therefore an approxi-

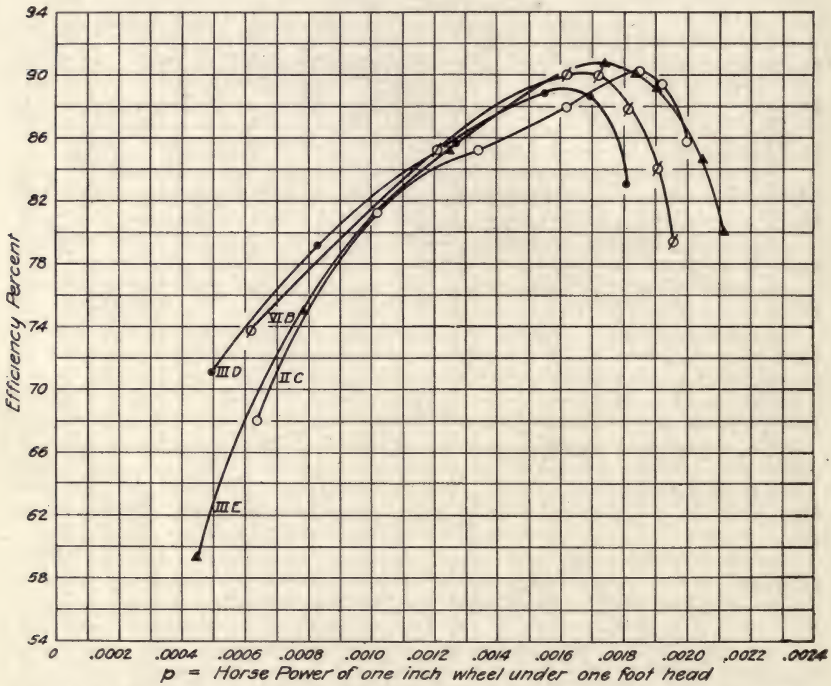

Fig. 249.—Power-Efficiency Curves for Four Wheels Which Give Over Eighty Per Cent. Efficiency With Specific Power of 3900 (see page 386).

mate analysis of the wheel or wheels probably best suited for the fixed conditions of any problem may be determined. It will be evident, however, that the graphical analyses described in this chapter are much better suited to determine these conditions and are the only available method by which a reasonably exact analysis can be made.

180. Practical Selection of Turbines.—The principles underlying the selection of the type of turbine necessary to fulfill a given condition, and the method of analysis to be applied thereto have already been discussed in sufficient detail. For the purpose of practice, it is

TABLE 37.

Table of Characteristic Coefficients of Various Wheels From Catalogue Values of Manufacturers.

Manufacturer	Type	ϕ_e	Δ_e	K_e	p_e	$\mathfrak{P}_e$
Christiana Mch. Co.	Balanced Gate Turbine	.668	1230	.00907	.000825	1250
Craig Ridgeway & Son Co....	Double Perfection...	.665	1225	.0125	.00115	1730
Craig Ridgeway & Son Co....	Standard	.668	1230	.00590	.000575	875
Barnard & Leas Mfg. Co......	McCormick's Holyoke	.665	1225	.0193	.00174	2625
DeHuff Eng. Co.	Burnham's New Improved Standard ..	.660	1215	.00933	.000868	1285
Platt Iron Works..............	Victor Cylinder Gate	.758	1397	.0249	.00225	4400
Platt Iron Works..............	Victor Wicket Gate	.832	1534	.0261	.00237	5550
Rodney Hunt Mch. Co........	Hunt McCormick ...	.907	1670	.0257	.00235	6575
Poole Eng. & Mach. Co.......	Poole-Leffel	.750	1380	.00627	.00063	1195
The Trump Mfg. Co...........	Standard Trump ...	.728	1340	.0256	.00234	4175
The Trump Mfg. Co...........	High Speed Trump..	.824	1518	.0334	.00304	7000

Characteristic Coefficients From Holyoke Tests.

Manufacturer	Holyoke Test	ϕ_e	Δ_e	K_e	p_e	$\mathfrak{P}_e$
Allis-Chalmers	2231	.678	1250	.00405	.000376	5900
Allis-Chalmers	2218	.804	1480	.0261	.00244	5350
Allis-Chalmers	2122	.822	1515	.0413	.00380	8740
Hydraulic Turbine Corp...............	2267	.727	1340	.0294	.00268	4810
J. & W. Jolly..........................	2141	.814	1500	.0246	.00229	5160
James Leffel & Co..................	2359	.901	1660	.0259	.00241	6650
James Leffel & Co..................	2363	.798	1470	.0527	.00488	10550
S. Morgan Smith Co................	2070	.732	1350	.01308	.00128	2330
S. Morgan Smith Co................	2263	.798	1470	.0300	.00289	6250
S. Morgan Smith Co................	2208	.760	1400	.0431	.00425	8330
Wellman-Seaver-Morgan Co.	2150	.705	1300	.00474	.00045	760
Wellman-Seaver-Morgan Co.	1799	.757	1395	.01935	.00182	3550
Wellman-Seaver-Morgan Co.	2270	.792	1460	.0360	.00344	7330
Wellman-Seaver-Morgan Co.	2281 2320	.754	1390	.0232	.00412	7960
I. P. Morris Co......................	2126	.689	1270	.0089	.00078	1260
I. P. Morris Co......................	2127	.798	1470	.02175	.00194	4200
I. P. Morris Co......................	2160	.786	1450	.0385	.00346	7270

desirable, however, to cover a wider range than is possible with the diagrams so far discussed. In order to permit of a considerable range in practice without the considerable labor involved in the preparation of a large number of graphical diagrams, a selection of six turbines has been made covering a wide range of specific power and graphical diagrams of the Holyoke test have been platted (see Figs. 250 to 255 inclusive). Each of these turbines represents a wheel made by a different manufacturer. The relations of efficiency to specific power in these six turbines are shown in Fig. 256, page 396, from which it will be seen that with these six wheels, specific power as low as $\mathfrak{P} = 500$ and as high as $\mathfrak{P} = 15,100$ may be obtained with efficiencies of not

less than eighty-five per cent.; while with a minimum limit of eighty per cent. efficiency, specific powers from $\mathfrak{P} = 300$ to $\mathfrak{P} = 16{,}500$ may be obtained. Graphical analyses of these wheels in the inverse order of their specific power, are shown in Figs. 250, 251, 252, 253, 254 and 255 (pages 390 to 395).

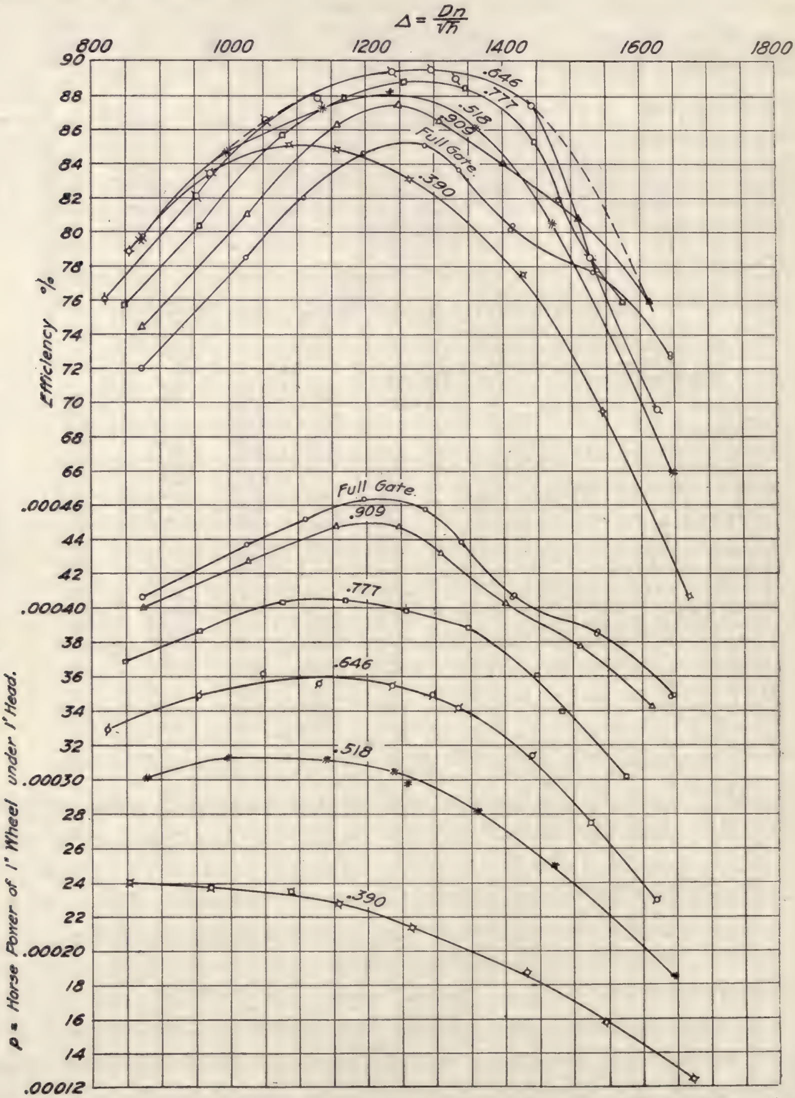

Fig. 250.—Analysis of Results of Holyoke Test Made on Wheel III A (see page 386).

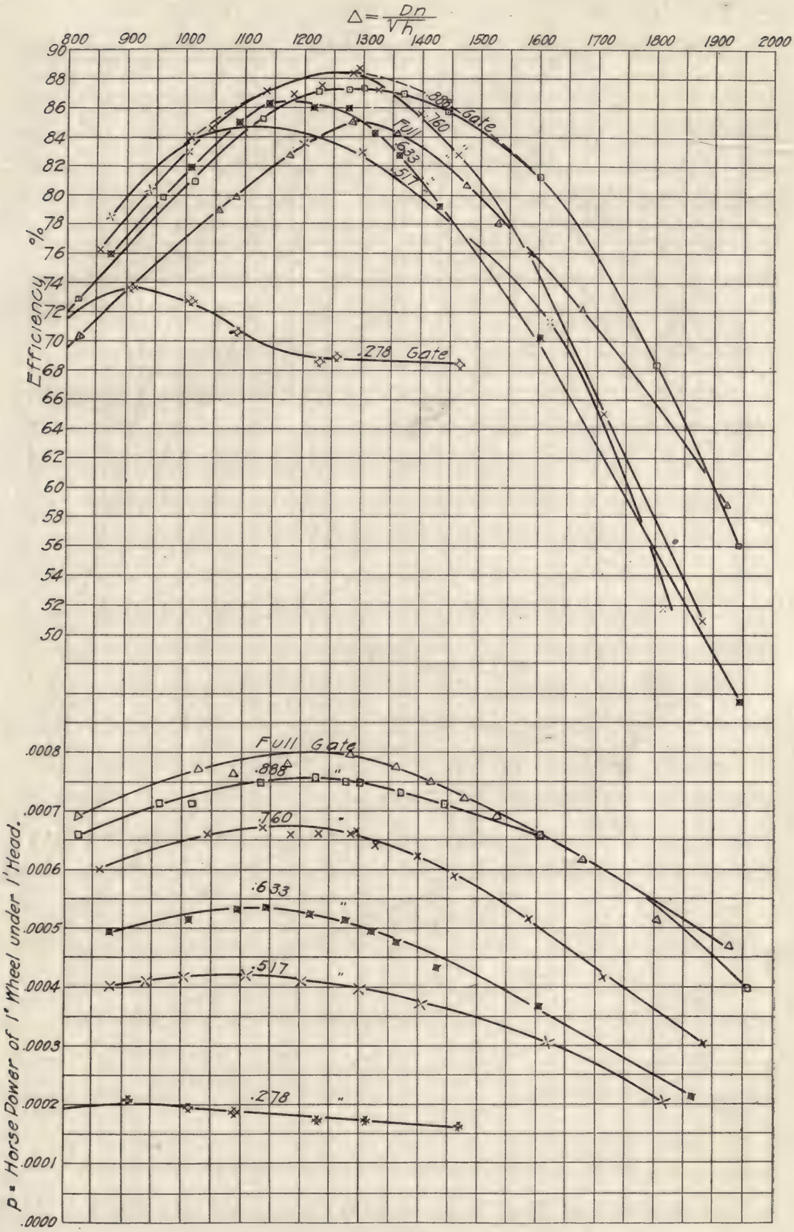

Fig. 251.—Analysis of Results of Holyoke Test Made on Wheel VI A (see page 386).

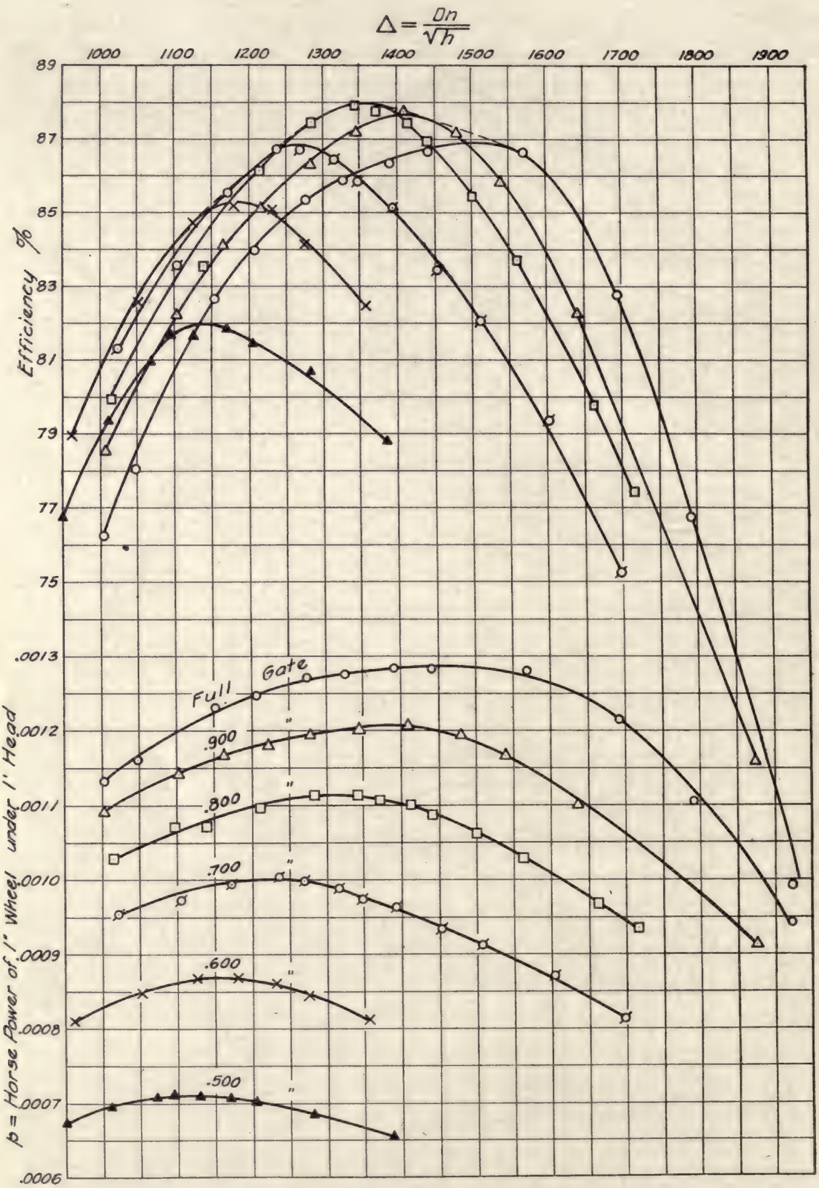

Fig. 252.—Analysis of Results of Holyoke Test Made on Wheel II B (see page 386).

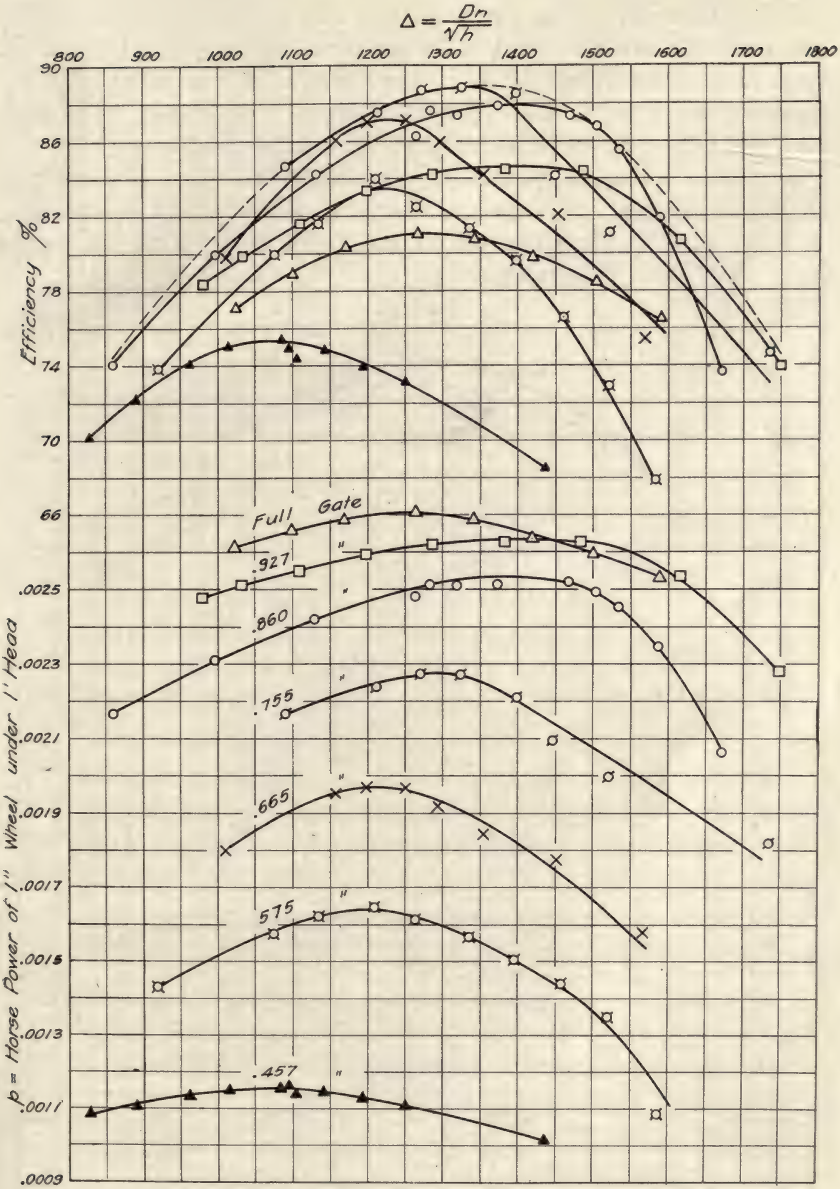

Fig. 253.—Analysis of Results of Holyoke Test Made on Wheel IV E (see page 386).

It will be noted in Fig. 256, page 396, that these e–𝔓 curves overlap and that with certain values of 𝔓 the same efficiency can be obtained from each of the wheels whose curves intersect at that point. Usually, however, the form of efficiency-power curves will so vary as to make one wheel the most satisfactory for certain fixed conditions.

For other values of 𝔓, the diagram shows that three of the wheels will be available at approximately the same efficiency, and in such cases analyses will also show that one of the wheels is best fitted for certain fixed conditions of operation.

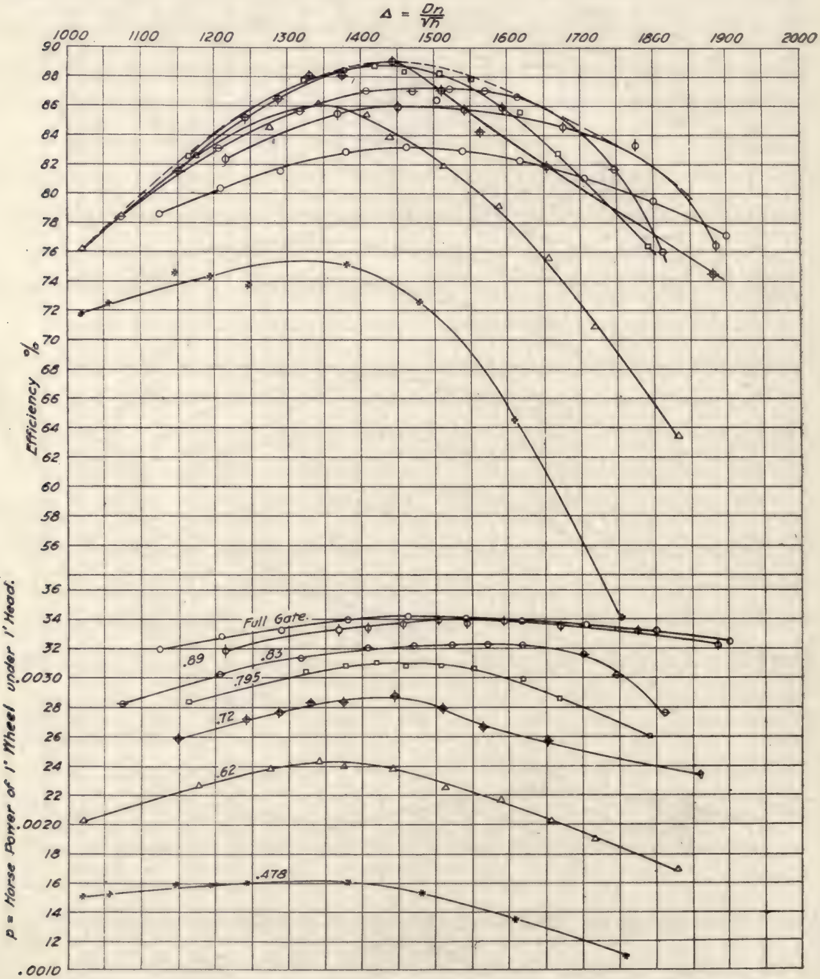

Fig. 254.—Analysis of Results of Holyoke Test Made on Wheel V D (see page 386).

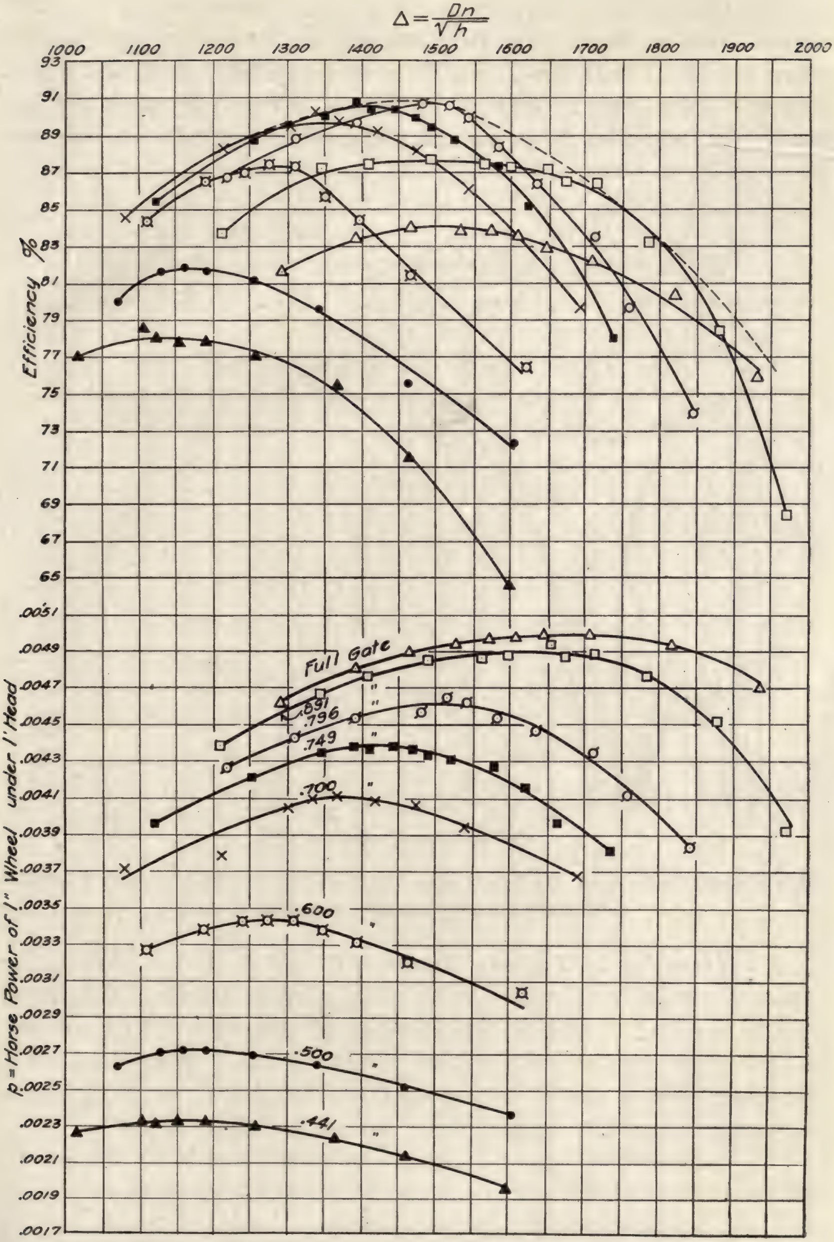

Fig. 255.—Analysis of Results of Holyoke Test Made on Wheel I B (see page 386).

With these various diagrams a wide range of problems can be quickly solved, with practice. It is desirable for the student to make one or more graphical analysis of wheels with practically the same specific power. These may be selected by means of Fig. 248, page 387, and the results should be compared with the results obtained from diagrams included in the text.

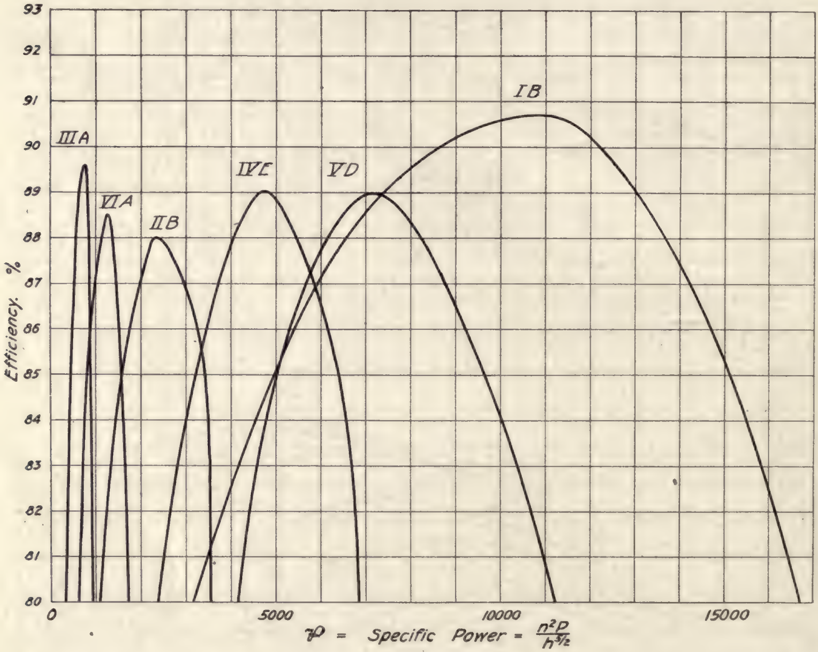

Fig. 256.—Relations of Efficiency and Specific Power for the Six Wheels of Different Manufacturers (see page 394).

181. **Necessity of Turbine Analyses.**—The tendency of manufacturers is always to furnish wheels of more than ample power, which usually involves a considerable sacrifice of efficiency. This tendency is encouraged by the fact that the capacity is easily determined, while the determination of efficiency in operation is always more difficult and frequently impossible on account of the setting of the wheel.

In a hydro-electric plant recently examined, where the wheels had not been analyzed and selected with competent engineering advice, but had been purchased on the recommendation of manufacturers, the following conditions (which illustrate the tendency above referred to) obtained. The units of the installation were 2,400 K. W. generators

directly connected to twin wheels rated at 4,000 H. P. at full load, and installed to operate the generators at about twenty-five per cent. overload. An analysis of the Holyoke test showed that under the head available the wheels would actually deliver more than 5,000 H. P. and were therefore much too large for efficient operation (see Fig. 257, page 398). This extra capacity would result in a net loss in efficiency of about five per cent. under the best commercial operating conditions. Through faulty operation, the generators were seldom allowed to operate at more than 1,500 H. P., hence improper selection and improper operation resulted in a net loss in efficiency of nearly sixteen per cent., which corresponds to a loss of about twenty-five per cent. of the water used. In this plant every cubic foot of the low water flow for six or eight months of each year was utilized and auxiliary power was necessary, so that the loss involved was serious and furnishes an important example of the necessities of careful and intelligent selection.

A smaller wheel installation and proper operation, or larger generators (3,000 K. W.), operated at an average load of seven-eighths their rated capacity would, with twenty-five per cent. overload capacity in reserve, give ample reserve capacity for sudden increases in power demand and would have resulted in a saving of twenty per cent. at no increase in expense. Such a saving often represents the difference between financial success and failure.

In the design of the wheel, it is seldom practicable to guarantee the power which will be developed by a wheel under a given head nearer than $\pm 2\frac{1}{2}\%$, and frequently a still greater variation will result. To illustrate this difficulty and also the necessity of analyses in order to determine the results which will be obtained for any particular wheel, the guarantees for a recent installation are compared with the actual test results secured by the manufacturer in Fig. 258, page 399). This figure shows the actual test results secured by the manufacturer with the first wheel, especially designed and constructed for the fixed conditions. The maximum efficiencies secured on the test were slightly above the efficiencies guaranteed but the power of the wheel was so great that the efficiencies under the ordinary range of working conditions were below the guarantee by about three per cent. This wheel was rejected, and a new wheel built by the manufacturer, which fulfilled the guarantee in a satisfactory manner (see Fig. 259, page 400).

182. Example of the Results of Turbine Analyses—Unsatisfactory Specifications.—It is highly essential that the specifications for turbines shall be so explicit in regard to the operating conditions that the

manufacturer will fully appreciate the full range of conditions under which the wheel is to operate. In an installation constructed some five years ago, it was desired to install a pair of wheels to operate a 1,000 K. W. generator under ranges of heads varying from sixty-five to eighty-five feet. The necessity of an exact detailed statement of the operating conditions was not appreciated at the time the specifications were prepared, although the necessity of accurate turbine analy-

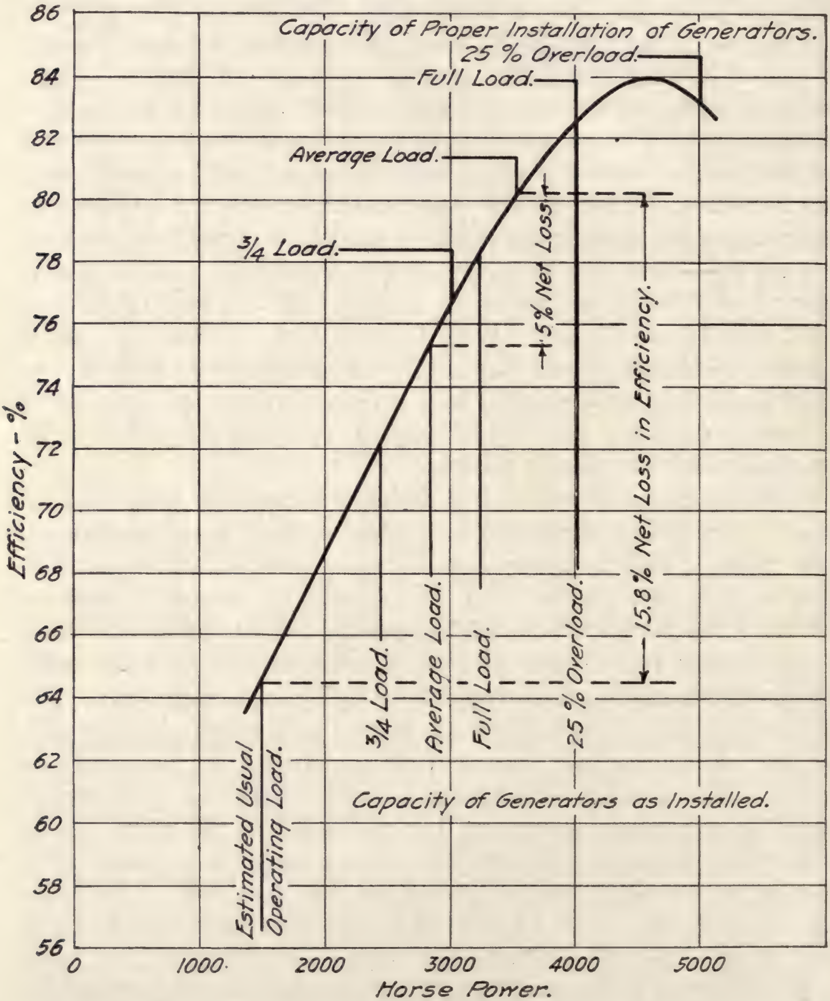

Fig. 257.—Efficiency-Power Curve for Wheel Too Large for the Requirements (see page 397).

ses and selection, in order to secure the best results, was fully understood. The specifications as prepared, read as follows:

"Each unit shall consist of a pair of turbines in tandem, and shall be capable of developing a maximum of 1,900 actual horse power under a working head of seventy feet when running at 375 revolutions per minute. These turbines are to be designed to operate satisfac-

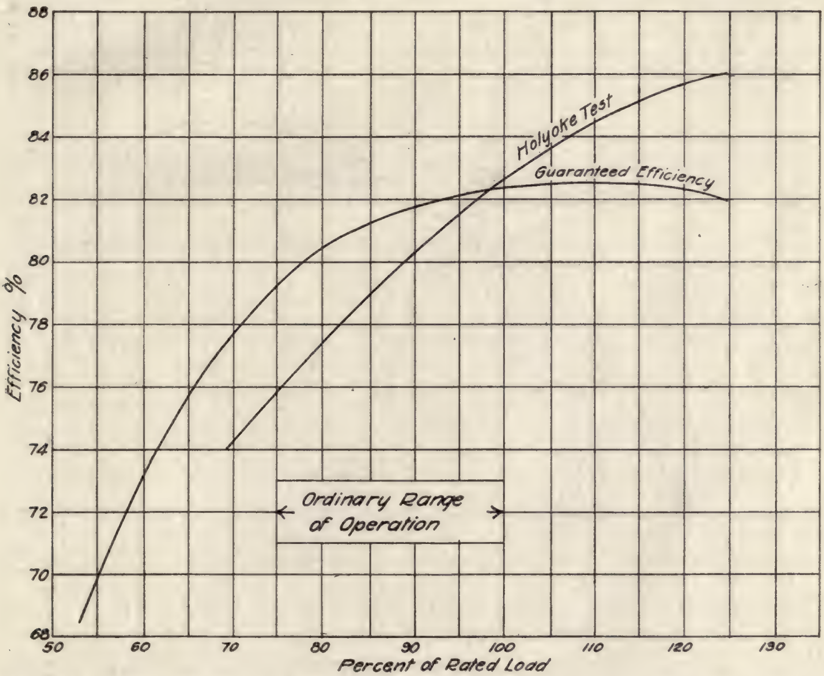

Fig. 258.—Comparison of Guaranteed Efficiency-Load Curve and the Results of Test of a Specially Designed Turbine (see page 397).

torily under a maximum head of eighty feet, and a minimum head of sixty-five feet if so required.

"The contractor shall furnish a Holyoke test sheet of a turbine of homogeneous design to that he proposes to furnish. Said test should be of a turbine of similar size to the turbines on which proposals are made, but, if such is not available, it may be on the nearest size available."

On the basis of these specifications, seven bids were submitted, and the guarantees offered by the contractor are shown in Fig. 260,* page

* See Trans. A. S. C. E., Vol. 64, p. 361.

401. The wheels actually submitted for this installation were in all cases—except in the case of wheel *A*—too large for the best results, and none of them would have fulfilled the guarantee as is clearly apparent from Fig. 261, page 402. For this reason, all bids were rejected and the matter was taken up directly with the manufacturer of wheel *A*, which came nearest to suiting the conditions of operation.

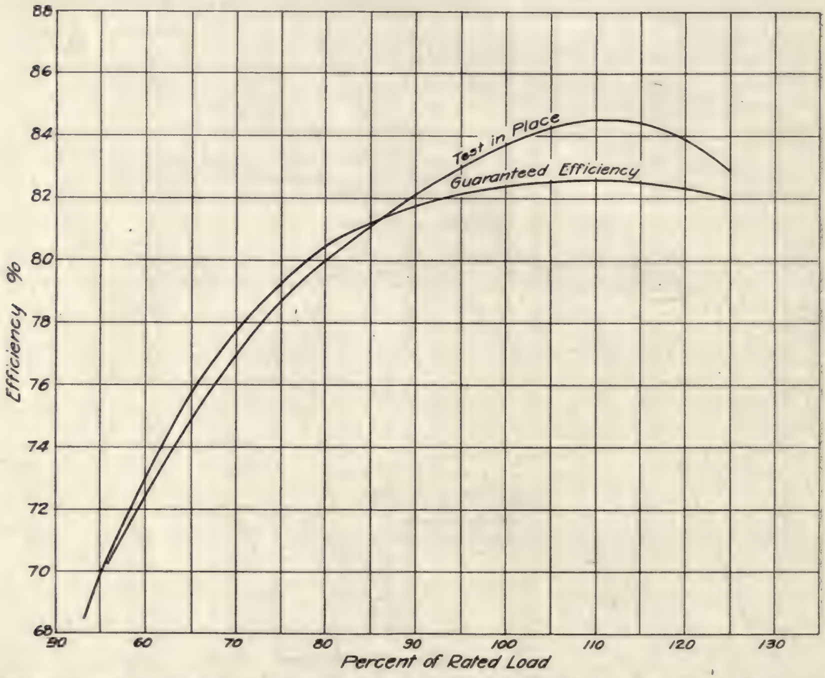

Fig. 259.—Efficiency-Load Curve for the Second Specially Designed Turbine (see page 397).

From a graphical analysis of a homologous turbine similar to wheel *A*, the efficiency power curves for the various ranges of heads were made as shown in Fig. 262, page 403. These curves show that a wheel of less diameter and of somewhat different characteristics, would best fulfill the conditions of operation, and a contract was entered into under which the efficiencies shown in Fig. 263, page 403, were guaranteed and afterwards essentially realized.

It may be noted that final surveys, completed before the letting of the contract, developed the fact that a maximum head of eighty-five feet could be obtained and that to secure the best results at all heads, less power than specified at the seventy foot head was desirable. Of

the wheels selected, five units were installed; with four of these units at full head the maximum designed capacity of the plant would be reached, and with five units the full capacity of the plant under the minimum head can be obtained.

It is evident from the bids received that the specifications were not sufficiently explicit, and while the desired results were assured before the final award, the specifications were clearly faulty.

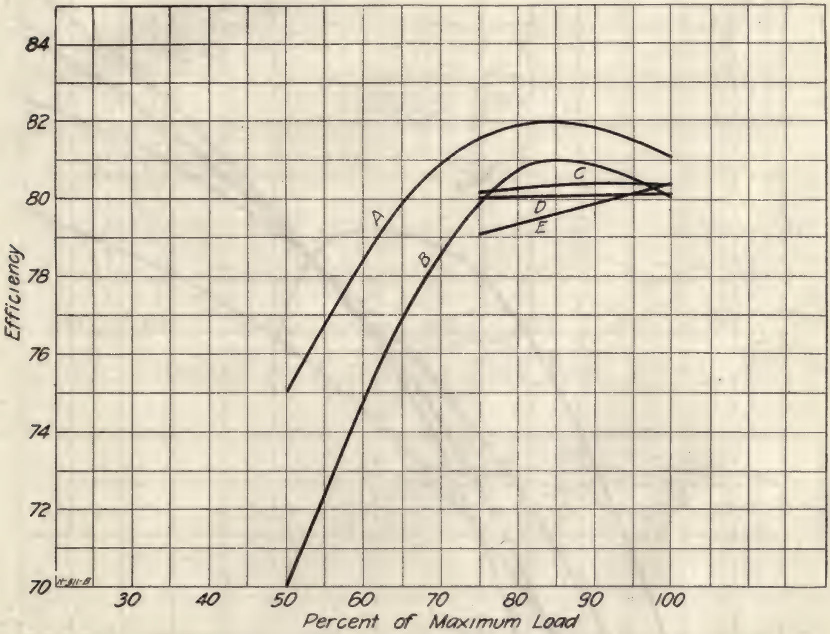

Fig. 260.—Guaranteed Efficiencies of Various Turbines (see page 399).

183. Explicit Specifications.—In a later installation, an attempt was made to remedy the defects previously discussed, and the conditions of operation of the desired turbines were specified as follows:

"Bids are desired on three units of two turbines each. Each unit shall consist of one pair of turbines mounted on a horizontal shaft. These turbines will be installed in open penstocks, and shall be capable of developing a maximum of about 650 actual horse power under a working head of sixteen feet, and running at approximately 128 revolutions per minute. These turbines are to be designed to operate satisfactorily under a maximum head of seventeen feet and a minimum of twelve feet, if so required. The efficiency of the units at each condi-

tion of load shall be not less than that guaranteed by the bidder in his proposal.

"Each turbine unit is to operate a sixty-cycle, three-phase alternator of 360 K. W. rated capacity at ninety per cent. power factor and ninety-four per cent. generator efficiency, and to a maximum of twenty-five per cent. overload.

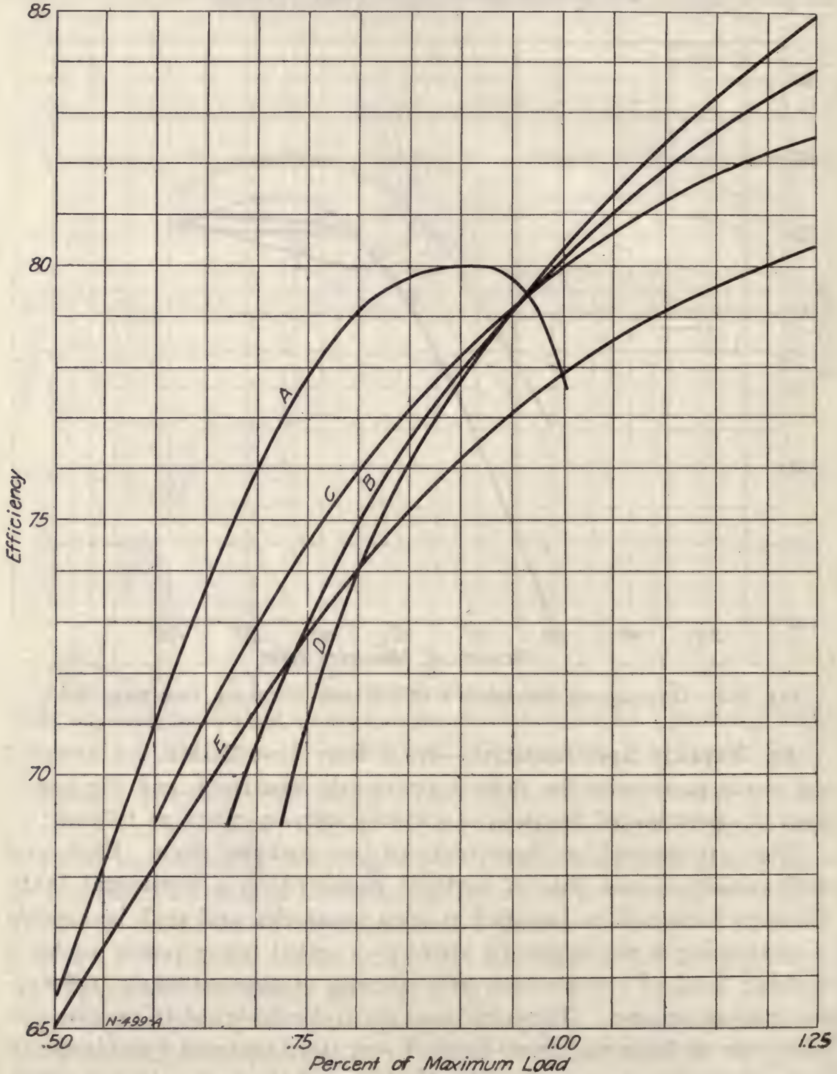

Fig. 261.—Efficiency-Load Curves from Holyoke Tests (see page 400).

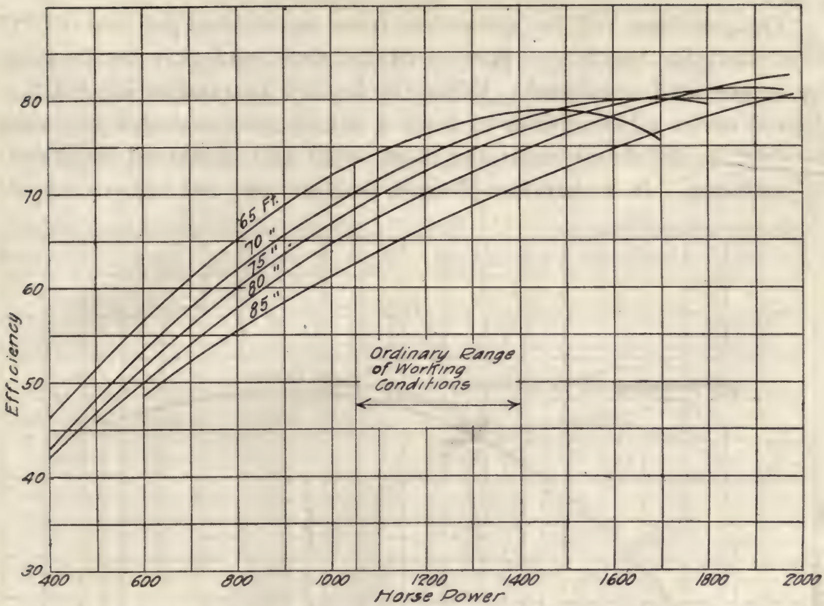

Fig. 262.—Efficiency-Power Curves of a Wheel Under Various Loads (see page 400).

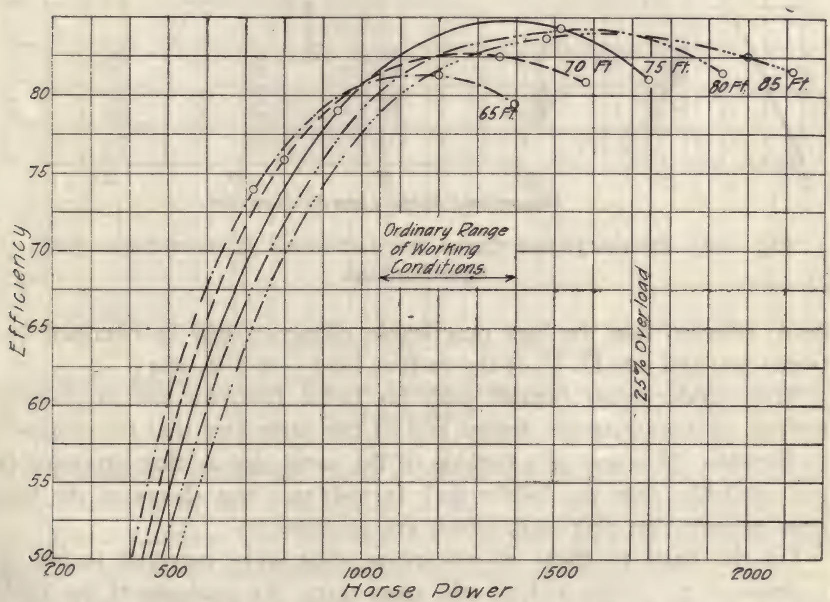

Fig. 263.—Guaranteed Efficiency-Load Curves for a Wheel Under Various Heads (see page 400).

"The generator will be operated at from seventy-five per cent. to full rated load for the larger portion of the time, and only occasionally under overload conditions. While the highest practicable efficiency is desired under all conditions of load, it is especially desirable to secure the highest efficiency under the most usual and continued conditions of operation. It is therefore desired that the size and type of wheels

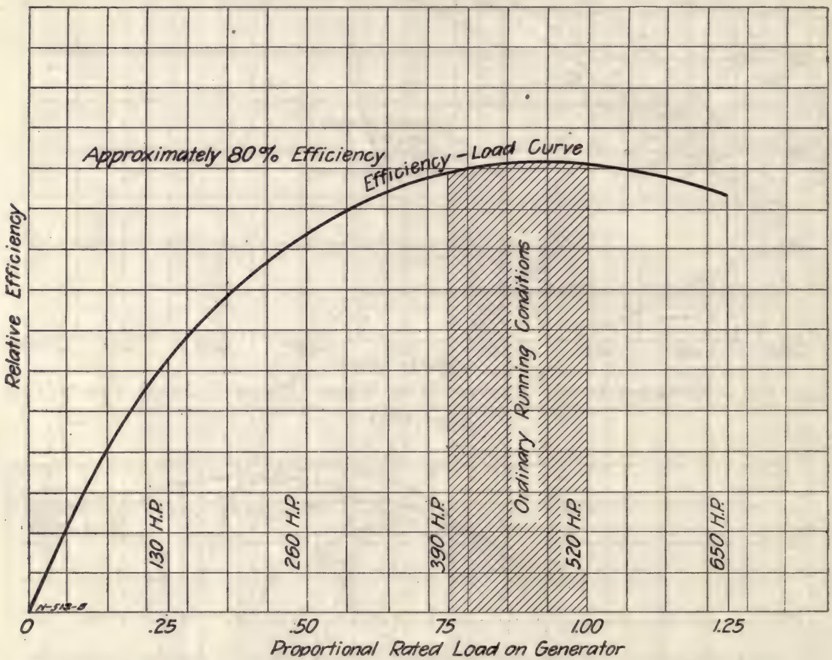

Fig. 264.—Efficiency-Load Curve for a Turbine in Accordance With Specifications.

be so selected that the best practicable efficiency will be obtained between 390 and 520 H. P. of the turbine load (see Fig. 264).

"The bidder must furnish with his bid a Holyoke test sheet of a turbine of homogeneous design and of the same size that he proposes to furnish. If a test of a turbine of the same size as that proposed is not available, then the bidder may furnish two test sheets of the two sizes nearest that proposed, which are available."

On the basis of these specifications, bids were received including guarantees as shown in Fig. 265, page 405. An analysis of the Holyoke tests of homologous wheels submitted with the bids showed,

however (see Fig. 266, page 406), that in only three cases, *viz.* 1, 2 and 4, the guarantees of the manufacturers were below the actual results which were shown by the Holyoke test.

It is apparent from the above discussion that the only way in which correct results can be obtained is for the engineer to appreciate the necessity and possess the ability to make (1) a comprehensive specification of the load requirements under the various conditions of head that

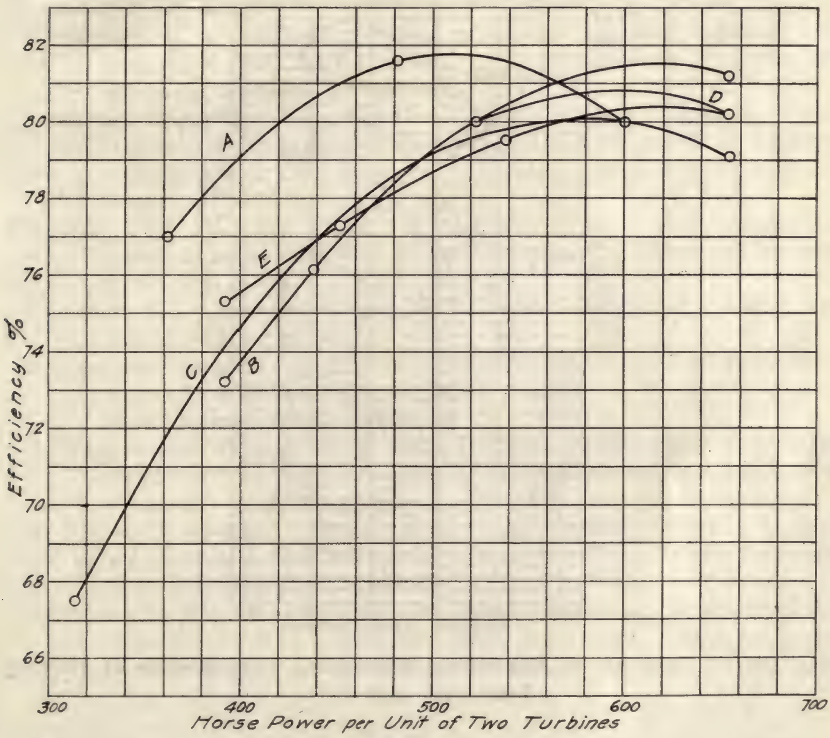

Fig. 265.—Efficiency-Load Curves Guaranteed for Various Turbines (see page 404).

will obtain and within the limits of physical possibility, and (2) a detailed analysis of the possibilities of the type of turbine proposed by the manufacturer. The engineer who neglects such an analysis, and blindly follows the recommendation and unverified guarantees of manufacturers, must expect to secure such unsatisfactory results as the contingencies of close competition and the temptation due to the possession of standards already developed, always create.

184. General Principles of Design and Selection.—It will be noted that in general the efficiencies of turbines increase materially with the load; it is therefore important to so design the plant and select the turbine that the installation may be operated as near full load (or most efficient load) as possible.

With quickly varying loads, which usually obtain in hydro-electric plants on general service, certain reserve capacity is essential. As

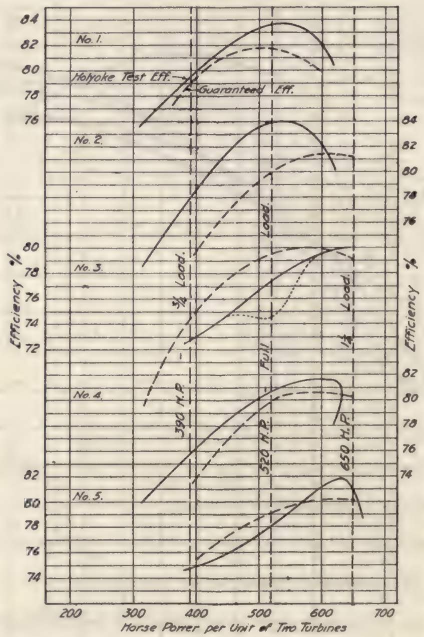

Fig. 266.—Comparison of Guaranteed Efficiencies With Results of Holyoke Tests (see page 405).

electrical generators can usually safely carry peak loads considerably in excess of their rated capacity for brief periods, the turbines to operate them must usually have a capacity sufficient to operate under overload conditions. This overload capacity should seldom exceed twenty-five per cent. of the rated capacity, otherwise low efficiencies will result. Where the load carried is fairly constant, little or no overload capacity is desirable as the better efficiency secured by operating constantly at or near the most efficient load will more than compensate for a reduction in the maximum power output, and a better efficiency

of one or two per cent. will usually more than compensate for the cost of an extra unit. As a general rule, a gain of five or six per cent. in efficiency is of greater value than the cost of an entire machine installation, and turbines which will give low efficiency are expensive at any price.

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CHAPTER XIV

THE SPEED REGULATION OF TURBINE WATER WHEELS

185. Symbols Used in This Chapter.—The following symbols will be used in the mathematical discussions which follow:

A = cross-sectional area of penstock in square feet.

$B = \dfrac{V + v_0}{V - v_0}$

b = length of brake arm.

c = friction coefficient for flow in pipe lines $= \dfrac{1}{2g}\left(1 + f\dfrac{1}{d} + \text{etc.}\right)$.

C = constant of integration.

D_a = maximum rise of water in standpipe above the forebay when full load ($v = v_f$) is rejected by the wheels.

D' = drop of water in standpipe below original friction gradient all influences considered.

D = ditto, friction in penstock neglected.

D_b = drop of level in standpipe below forebay.

d = diameter of penstock (closed circular) in feet.

$e = 2.71828$ = base of natural system of logarithms.

E = efficiency.

F = cross-sectional area of the standpipe in square feet.

f = "friction factor" in penstock.

g = acceleration due to gravity in feet per second per second.

H = total available power head in feet.

H' = effective head at the wheel $= H - h_F$ for any given uniform velocity, V, in the penstock.

h = instantaneous effective head at the wheel during changes of velocity in the penstock.

h_a = head which is effective at any instant in accelerating the water in the penstock and draft tube.

h_F = friction loss in penstock for normal flow with a given head and gate opening.

h_f = variable head lost by friction entrance, etc., in penstock when the velocity is v.

I = moment of inertia or fly wheel effect of revolving parts in pounds at one foot radius = foot^2 pounds.

K = energy delivered to the wheel.

$\triangle K$ = excess or deficient energy delivered to wheel during change of load.

$\triangle K_1$ = excess or deficient energy delivered to wheel due to excess or deficiency in quantity of water during load change.

$\triangle K_2$ = ditto,—due to energy required to accelerate or retard the water in the penstock.

$\triangle K_3$ = ditto,—due to sluggishness of gate movement.

K' = kinetic energy in foot pounds of revolving parts at speed S.

$\triangle K'$ = increment (+ or —) in K' due to load change.

$$k = \frac{2gH}{lV}$$

$$k' = \frac{2gH}{2.31V}$$

l = length of penstock in feet.

M = slope of the $v-t$ curve when $v = \dfrac{v_0 + v_1}{2}$ (equation 150).

P = horse power.

p_0 = initial horse power output from the water wheel.

p_1 = the horse power output from the water wheel corresponding to the new load.

Q = discharge of the wheel under normal effective head H' for any given load.

q = instantaneous discharge of wheel in cubic feet per second during load change.

R = ratio of actual deficient or excess work done on wheel to that computed.

n = revolutions per minute.

S = normal R. P. M. of the wheel and other rotating parts.

$\triangle S = S - S_1$ = temporary change in speed.

S_1 = speed in revolutions per minute after load change.

T' = approximate time required for acceleration or retarding of water from velocity v_0 to v_1.

T'' = the time required for the governor to adjust the gate after a change of load.

t = variable time after gate movement.

V = normal (and hence maximum possible) velocity in the penstock with given head and gate opening.

v = instantaneous variable velocity in the penstock while adjusting to a new value.

v_0 = velocity in penstock at the instant of gate change.

v_1 = velocity in the penstock required for new load.

W = weight on scale pan.

w = weight of a cubic unit of water in pounds.

Y = maximum departure of head h, from normal with use of stand-pipe,—discharge of wheel assumed constant at the abnormal head (see D_a and D_b).

y = variation of water level in the standpipe from forebay level $= H - h$.

δ = speed regulation or per cent. variation of speed from normal.

λ = velocity of wave propogation feet per second.

186. The Relation of Resistance and Speed.—The power delivered by any water wheel may be expressed in terms of resistance over-

come by the wheel through a known distance and in a known time by the formula

$$(132) \qquad P = \frac{2\pi \, b \, W \, n}{33000}$$

The second term of this equation may be divided into two factors: first,

$$\frac{2\pi \, b \, W}{33000}$$

which may be called the resistance factor and which is the resistance overcome or power produced by the wheel per revolution per minute; and n, the number of revolutions per minute. The product is the horse power of the wheel.

In any wheel operating with a fixed gate opening and under a fixed head the speed n, will always increase as the resistance w, decreases, and will decrease as the resistance increases.

In Fig. 267, page 411, the line AB shows the relation of speed to resistance in a turbine operated with a single fixed gate opening and for the full range of load conditions (as determined by experiment) from A, at which the resistance w, was so great as to hold the motor stationary, to B where the resistance was completely removed and the entire energy of the applied water was expended in overcoming the friction of the wheel, or rejected as velocity energy in the water discharged therefrom. From this figure it is evident that if, at any fixed gate opening, a wheel is revolving at a given speed n, and the resistance w, is decreased to w'' the speed will increase to n'', while if the resistance increases to w' the speed will decrease to n'.

187. Self-Regulation in a Plant With Variable Speed and Resistance.—At Connorsville, Indiana, is a pumping plant (Fig. 268) in which a single horizontal shaft turbine is directly connected through friction clutches to two rotary pumps. For operation the turbine gates are opened until the pump, or pumps, speeding up to a suitable R. P. M., produces the desired pressure in the distributing system. The work of the pump under these conditions in pumping water at the speed of operation against the desired pressure equals the work done by the quantity of water q, passing through the turbine, less friction and other losses. If the pressure falls, the loads become unbalanced: i. e., the resistance is reduced and the turbine and pump increase in speed until the balance is restored. If the pressure rises the machine slows down until there is again a restoration of balance between the power of the turbine, the pump load and friction losses.

To pump water against an increased pressure, it is necessary to increase the gate opening of the turbine. In its regular daily work the varying demand for water is thus supplied by the self-regulation of the two machines used and no governor is needed. The conditions of operation are similar to those illustrated in Fig. 267.

188. The Relations Necessary for Constant Speed.—Fig. 269, page 412, is a diagram drawn from experimental or test observations and similar to Fig. 267, except that the relations between speed and resistance are shown for various gate openings.

Fig. 267.—Relation of Resistance to Speed.

It is evident that if the turbine must operate at a fixed speed n, and the resistance w increases to w' or decreases to w'', it will be necessary to increase the gate opening from seven-eighths gate to full gate in the first case and to decrease it to three-fourths gate in the second case in order to maintain a uniform speed.

An examination of the load curves described in Chapter III shows that changes in load are constantly in progress. For the satisfactory operation of water wheels, under these constant and irregular changes in load, automatic regulation of the turbine gates becomes necessary. This is accomplished through the water wheel governor which regulates the gates through the various classes of gate mechanisms described in Chapter X.

The power output of a water turbine in terms of energy applied to the wheel is expressed by the formula—

$$P = \frac{QH'E}{8.8} \qquad (133)$$

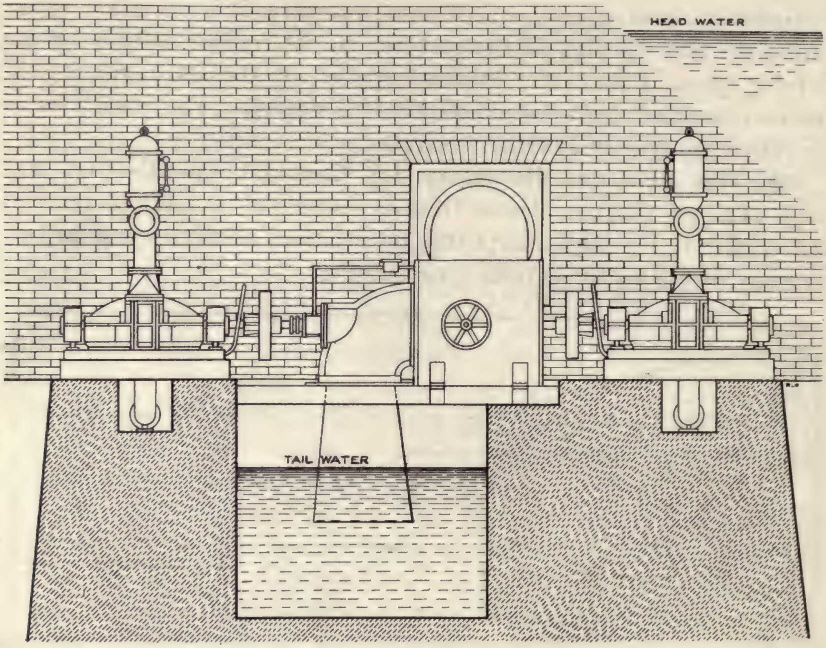

Fig. 268.—Connorsville Turbine Driven Pumping Plant, Partially Self-Regulating (see page 410).

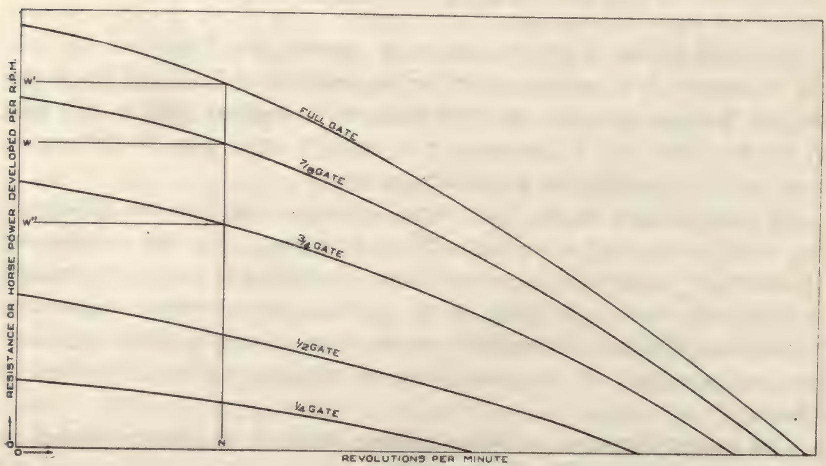

Fig. 269.—Relations of Resistance and Speed at Different Gate Openings (see page 411).

Any sudden increase or decrease of load w, will produce a corresponding decrease or increase, respectively, in the speed n, of the machine as shown by Fig. 267, page 411, unless the energy applied to the turbine is immediately changed to correspond. The ideal turbine governor would effect a change in output by varying only q, thus obtaining perfect water economy by conserving unneeded water for future use. This is not possible in practice as head, water, and therefore efficiency are usually wasted when operating a wheel under other than its normal load and during the change in load.

189. Present Status.—The success of the application of hydraulic power to the operation of alternators in parallel and to the generation of current for electric lighting street railway and synchronous motor loads has been largely dependent on obtaining close speed regulation of the generating units accompanied with good water economy and without undue shock upon machinery and penstocks while working under extremely variable loads.

The degree of success thus far obtained in the development (necessitated by the above conditions) of automatic turbine governors, although achieved from the experimental standpoint almost exclusively, has been remarkable. Instances are now by no means uncommon where hydro-electric units working upon variable loads are controlled as satisfactorily as modern steam driven units. To accomplish this result the conditions must be especially favorable.

Success in this feature of hydro-electric design is by no means uniform, however, and the frequent failure to realize satisfactory results is often due to the lack of proper consideration of the arrangement of the mechanical, hydraulic, and electrical elements of the plant and generators, rather than to any inherent defects in the governor itself. The power plant, the turbines, the generators, and the governors are commonly designed by four different parties without proper correlation of study and design. At present neither experimental data nor theoretical formula are available by which the hydro-electric engineer can design his plant for an assumed speed regulation, or can predetermine the speed regulation which is possible with a given installation or the time required for the return to normal speed,—and yet the governor builder is commonly required by the engineer to guarantee these operating results. The predetermination of speed variations during portions of the steam cycle and at load changes has received careful study in the design of reciprocating steam engines and the desirable per cent. of speed regulation is freely guaranteed and readily obtained

through careful study and analysis by the designer. The same amount of study is warranted but seldom or never given to the problem of speed regulation in water power work.

190. Value of Uniform Speed.—Uniform, or nearly uniform speed is of great economic value in the operation of a plant but adds to the first cost and may also result in a waste of water. The correct solution of any given problem of speed regulation involves a compromise between first cost, water economy and speed regulation.

A pecuniary value cannot well be placed upon good speed regulation. It differs from poor speed regulation chiefly in procuring a more satisfactory operation of motor driven machinery and in producing a more constant incandescent light. Fluctuations in the brightness of a light are annoying, and tend to create dissatisfaction among consumers. Fig. 270 shows the general way in which the candle power of an incandescent light varies with the impressed voltage.* A pressure variation of five per cent., and hence also a speed variation of a similar amount, is shown to produce a much larger variation in candle power of the light,—in this case about twenty-five to thirty per cent.

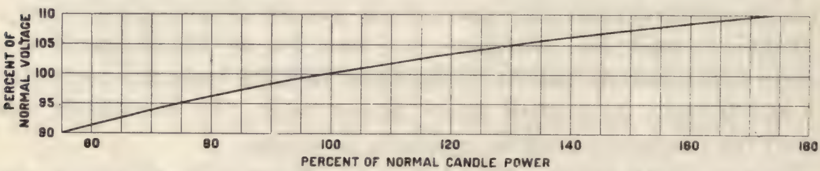

Fig. 270.—Relation of Normal Candle Power of Incandescent Lamps to Normal Voltage of Line.

Figures 271 and 272, pp. 415, 416, show the actual voltage and candle power fluctuations on the lines of two operating plants.* Fig. 271 shows poor regulation and Fig. 272 shows fluctuations due more particularly to line drop, the line voltage decreasing as the load increases.

191. The Problem.—Where (as in Fig 268, page 412) a turbine is operating under balanced conditions and the resistance changes in magnitude, the turbine does not at once assume the new speed relations corresponding to the change in resistance. The inertia of the moving parts of the wheel and of the column of water in the penstock, turbine and draft tube, tends to maintain uniformity of speed, and the wheel gradually changes in speed to that corresponding to the new conditions. In such cases the speed of operation is not essential and the

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* See "Voltage Regulation and Illumination" by Prof. L. B. Spinney. Elec. World, May 5, 1910, page 1151.

delay in reaching the speed corresponding to the resistance or work the turbine must perform is usually unimportant.

When, as in Fig. 269, page 412, the wheel is designed to operate at a fixed speed, the uniformity of speed becomes a matter of greater or less importance depending on the character of the work the wheel is to perform. In this case the inertia of the wheel and of all rotating parts of other machinery connected thereto tends to maintain a constant speed. On the other hand, the flow of water in penstock, tur-

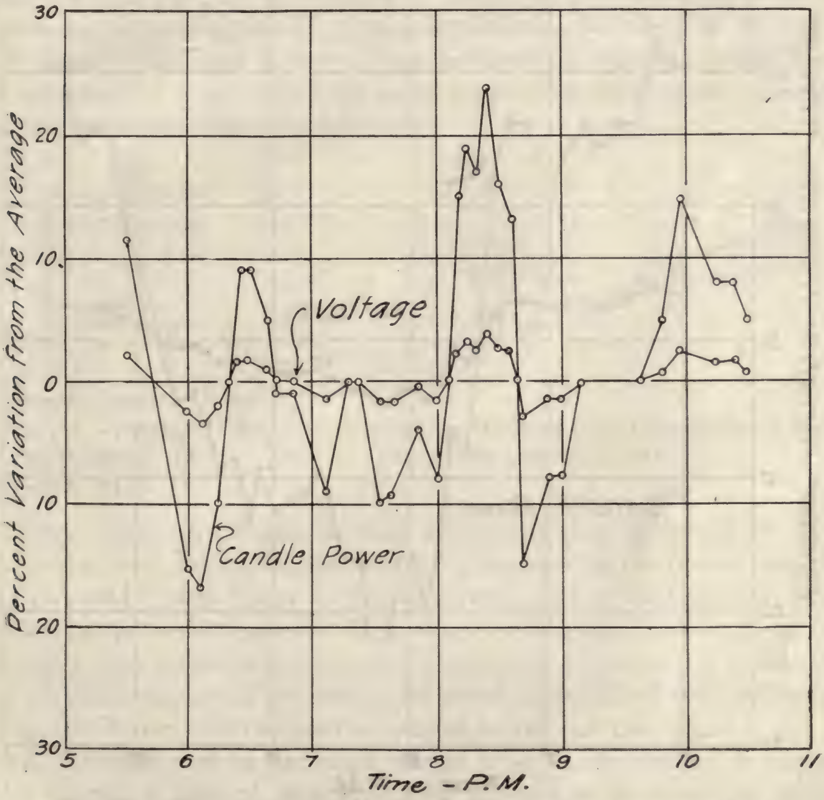

Fig. 271.—Voltage and Candle Power Fluctuation (see page 414).

bine, and draft tube must be changed in quantity (see equation 133), hence in velocity, and its inertia therefore tends to produce a change in head and to produce effects opposite to those desired for efficient regulation.

The conditions of installation have a marked effect on the difficulties of turbine governing. If (Fig. 273, page 417) the turbine is installed in an open pit and has only a short draft tube, and the water flows to the

gates from every direction, the velocity of flow from all directions is very low. The quantity of water which moves at a high velocity is confined to that in the wheel and draft tube and the change in the velocity and momentum, due to a change in the gates, produces no serious effects. If, however, water is conducted to and from the wheel

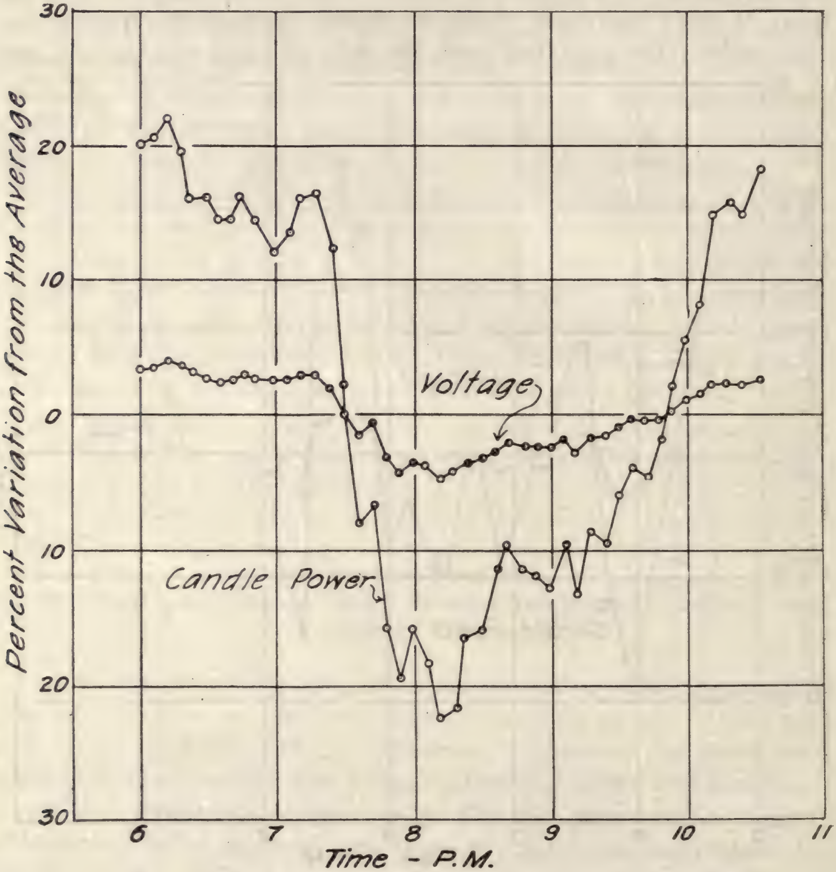

Fig. 272.—Voltage and Candle Power Fluctuation (see page 414).

through a long penstock and draft tube (as illustrated by Fig. 274, page 417) the conditions become quite different. In this case a large amount of energy is stored in the moving column of water and a change in its velocity involves a change in its kinetic energy which may, if an attempt is made at too rapid regulation, leave the wheel deficient in energy when increased power is desired, or, when the power

is decreased, may produce such shocks as will seriously affect regulation or perhaps result in serious injury to the penstock and wheel.

192. Energy Required to Change the Penstock Velocity.—An increase or decrease of load requires an ultimate increase or decrease in velocity of the water in the penstock. Work has to be done upon the water to accelerate it and must be absorbed in order to retard it. The total available power which can be expended for all purposes at any instant during the acceleration is (since vH is proportional to qH) proportional to the product of the instantaneous velocity and the supply head. This total power is thus definitely limited and, hence, *the work required to accelerate the water must be obtained at the expense of the work done upon the wheel.*

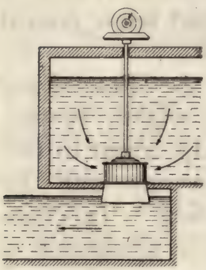

Fig. 273.—Turbine Installation in Open Pit.

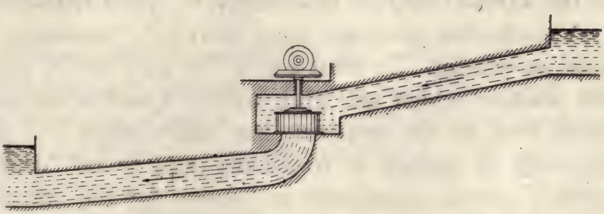

Fig. 274.—Turbine Installation With Long Penstock and Draft Tube (see page 416).

Thus, when an increase of load occurs the gate is opened by the governor, and the immediate result is a decrease in the power output of the wheel, even below its original value, and is diametrically opposed to the result desired. This counter effect may last for several seconds, and, unless sufficient reserve energy in some form is available to partially supply this deficiency, the speed of the wheel may fall considerably before readjustment to normal power can take place.

In the same way an excess of energy must be absorbed to decrease the velocity at time of decreasing load. This may be expended upon the wheel thus increasing the speed above normal, or it may be dissipated in one of several ways to be discussed later.

The water in the draft tube must be accelerated and retarded at each change of gate opening and its kinetic energy changed at the expense of the power output in exactly the same manner as that in the penstock. For this reason it should be included in all calculations as a part of the penstock. One additional precaution must be taken: if the draft head is large a quick closure of the turbine gate may cause

the water in the draft tube to run away from the wheel (actually creating a vacuum in the draft tube) and then return again causing a destructive blow against the wheel.

193. **Hunting or Racing.**—The regulation of both steam engines and hydraulic turbines as now accomplished is one of degree only since a departure from normal speed is necessary before the governor can act. Since the immediate effect of the gate motion is opposite to that intended, the speed will depart still further from the normal. This tends to cause the governor to move the gate too far with the result that the speed will not only return to normal as soon as the inertia of the water and of the rotating parts is overcome, but may rush far beyond normal in the opposite direction. The obvious tendency is thus to cause the speed to oscillate above and below normal to the almost complete destruction of speed regulation.

A successful governor must therefore "anticipate" the effect of any gate movement. It must move the gate to, or only slightly beyond, the position which will give normal speed when readjustment to uniform flow in the penstock has taken place. A governor with this property or quality is commonly said to be "dead-beat." In Chapter XV several expedients are shown for the automatic elimination of excessive racing.

194. **Shock or Water Hammer Due to Sudden Changes in Velocity.**—The acceleration or retardation of a moving body requires an unbalanced force. Since acceleration and retardation are identical, except as to sign, the required accelerating force may in all cases be expressed as follows:

$$\text{Force} = \text{mass} \times \text{acceleration.}$$

Acceleration, or the rate at which the velocity increment increases per increment of time, is expressed by the formula:

$$\text{Acceleration} = \frac{dv}{dt} \qquad (134)$$

The mass of water to be accelerated is

$$\text{Mass} = \frac{Alw}{g} \qquad (135)$$

Figures 275 and 276, page 419, show the conditions existing during an increase and decrease of velocity respectively. If the draft tube were closed at the lower end and no water leaving, there would be a total force, equal to the hydraulic pressure over the area of the penstock, or wAH, tending to move the water.

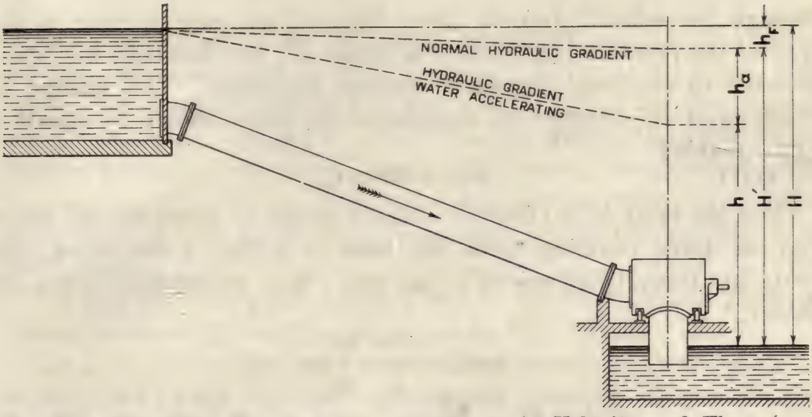

Fig. 275.—Condition Existing During Increase in Velocity and Flow (see page 418).

If the water is flowing with a velocity v, the turbine offers a resistance to flow represented by the effective head h, at the wheel, and the penstock offers a resisting head h_F composed of friction, entrance, and other losses. If the velocity remains uniform, $h = H'$, and the forces are balanced thus:

$$H = H' + h_F \tag{136}$$

If the opening of the turbine gate is now suddenly increased, the head H' at the wheel, will fall to the value h (shown in Fig. 275, above) which is required to force the given amount of water Av,

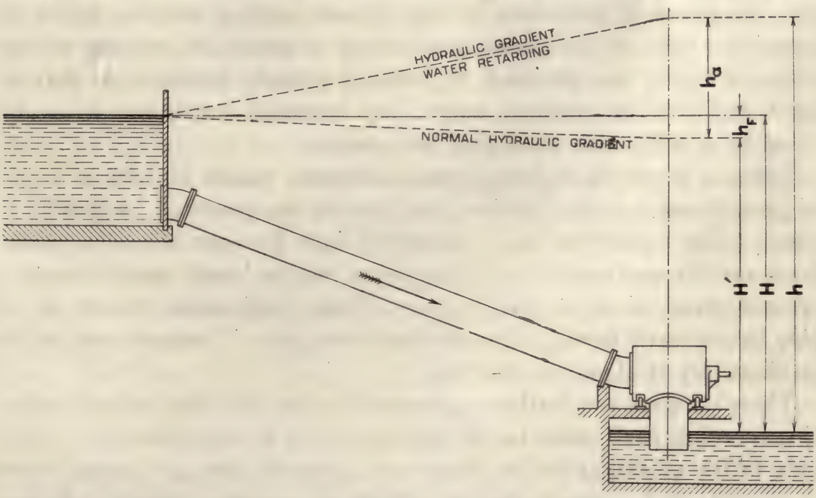

Fig. 276.—Condition Existing During Decrease in Velocity and Flow (see page 418).

through the wheel. On the other hand, if the gate opening is decreased the pressure head must rise above H' (as shown in Fig. 276) in order to discharge the water through the wheel. This change h_a in the head H' disturbs the equilibrium of forces shown by equation (136) making

$$h_a = h - H' + h_F \tag{137}$$

Only the head h_a is effective in accelerating or retarding the water and the force resulting from this head is wAh_a. Substituting this value and those of equations (134) and (135) in equation (133) we obtain:

$$wAh_a = \frac{Alw}{g} \cdot \frac{dv}{dt} \text{ or}$$

$$h_a = \frac{1}{g} \cdot \frac{dv}{dt} = \frac{1}{g} \times \text{(rate of velocity change)} \tag{138}$$

The value of h_a given by formula (138) is a general expression for the change in pressure-head due to a change of velocity or for the head which must be impressed to produce a desired change in velocity. When in excess of the static pressure as shown in Fig. 276, page 419, it is commonly called "water hammer" (see Section 212).

If the closure of the gates is rapid the value of h_a is large and the column of water is set into vibration or oscillation. If the partial closure of gate is sufficiently slow to allow a distribution of each increment of pressure along the pipe, this oscillatory wave is avoided and the pressure produced at any instant during closure, given by equation (138), is that which is necessary to retard the moving column of water at the rate at which its velocity actually decreases at that instant and can be reduced below any assumed maximum allowable value by a sufficiently slow gate movement.

When a penstock is long, these oscillatory waves become a source of great danger to the turbines and also to the penstock, especially at bends. The extinction of a velocity of four feet per second at a uniform rate in one second in a pipe 1,600 feet in length would create a pressure-head of about 200 feet, or a total longitudinal thrust on the pipe line at each bend, and upon the wheel gate, if twenty-four inches in diameter, of about twenty tons.

These dangers are further augmented by the fact that several waves, if succeeding each other by an interval which is approximately a multiple of the vibration period of the pipe, may pile up, so to speak, crest upon crest and cause a pressure which no possible strength of parts could withstand.

195. Permissible Rate of Gate Movement.—Gate movements must be sufficiently slow to avoid oscillatory waves of dangerous amplitude. No general quantitative rule can be given for the required rate of movement. It can be more rapid the shorter the penstock and the smaller the velocity in the same. The danger is much smaller during opening than during closure of a gate and the rate of gate movement could well be made much more rapid in the former than in the latter case.

The rapidity with which a gate should be opened is limited for feeder pipes with an initial flat slope as shown in Fig. 277, page 422.

Let h' be the lowest head obtained in opening the gate at an assumed rate and AB, the resulting hydraulic gradient. In case the gate opens so rapidly as to cause the distance a, at any point along the pipe to exceed suction limit, the water column in the penstock will separate (the portion of the column above A not being able to accelerate as rapidly as that below) and will again reunite with a severe hammer blow. Failure to observe this precaution probably caused the destruction of the feeder pipe of the Fresno, California, power plant. The rate to be used can be chosen after a determination, by the method discussed in Section 212, of the pressures resulting from several assumed rates of movement. The method is tedious but justifiable in many cases.

196. Regulation of Impulse Wheels.—It is impracticable, if not impossible, to build a pipe line strong enough and well enough anchored at all points to withstand the enormous pressures and longitudinal thrusts which would result from rapid gate closures in a long closed penstock such as commonly used for impulse wheels. The adjustment of quantity q, for changes in load of short duration is hence impossible in such closed penstocks and the expedient usually adopted is to "deflect" the jet from the wheel by changing the direction of discharge of a pivoted nozzle. This requires that the "needle valve" (see Fig. 183, page 273) or gate maintain a jet sufficient to carry peak loads; hence causing a waste of water at all other times. This condition is commonly improved somewhat by adjusting the valve about once each hour by means of a slow motion hand wheel for the maximum peak load liable to occur during that hour.

An automatic governor has recently been invented which moves the needle valve or gate slowly, thus adjusting for changes of load of long duration while it still retains the deflector to provide for abrupt changes in the load curve (see Fig. 286, page 460).

Another device proposed for use in this connection is a by-pass nozzle arranged to open as the needle valve rapidly closes, and then automatically close again at a rate sufficiently slow to reduce the excess pressure to safe limits. One advantage in favor of this arrangement is that the jet would then always strike the center of the buckets which is found to considerably reduce their wear.

An automatic relief valve of hydraulic or spring type is nearly always used but serves more as an emergency valve to reduce water hammer pressures than as a by-pass to divert water from the wheel for the purpose of governing. For this latter use the spring type of valve has proven unsatisfactory.

In some cases the water discharged from high head plants is used below for irrigation and must be kept constant, thus doing away with the necessity of varying the velocity in the feeder pipe for a varying load.

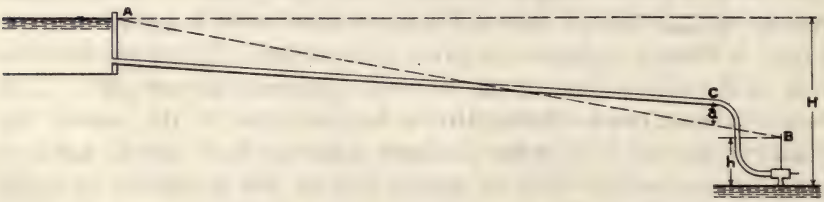

Fig. 277.—Feeder Pipe Having Initial Flat Slope (see page 421).

Mr. Raymond D. Johnson proposes for these high head plants, the use of large air chambers or "Surge Tanks," placed near the wheels, of a sufficient size so that the governor can control the needle valve directly, thus dispensing with the deflector and by-pass and doing away completely with the waste of water occasioned by their use. He has derived formulas by which he claims to accurately proportion these tanks for an assumed maximum allowable range of head fluctuation or surge.*

197. **Influences Opposing Speed Regulation.**—Abrupt changes in the demand for power of a considerable proportion of the total capacity of a plant, take place at times in modern power plants. Three causes tend to make the change in output of a wheel lag behind the change in demand placed upon it; viz.: (1) the fact that the governor, however sensitive, does not act until an appreciable change of speed occurs, and then not instantly; (2) the fact that some time is required

* See "The Surge Tank in Water Power Plants," by R. D. Johnson. Trans. Am. Soc. M. E., 1908.

for the readjustment of penstock velocity, even after the gate movement is complete; (3) the necessity of changing the velocity, and hence of overcoming the inertia of the water in the penstock and draft tube at each change of load.

Each of these influences is directly opposed to speed regulation, as will appear in the succeeding articles, since each causes the power supplied to a wheel, at time of increasing load, to fall short of the demand, the deficiency being supplied at the expense of the speed from the kinetic energy stored in the rotating parts. The expression for the total deficient work, i. e. foot pounds, is:

$$\triangle K = \triangle K_1 + \triangle K_2 + \triangle K_3 \tag{139}$$

for which see equations (153) and (155) and Sections 199 and 201.

198. Change of Penstock Velocity.—Assuming the gate movement to take place instantly, we will have the condition illustrated in Figures 275 or 276, page 419, for which equation (138) was derived (see Section 194). Solving equation (138) for $\frac{dv}{dt}$ we have:

$$\text{Acceleration} = \frac{dv}{dt} = \frac{g}{l} \times (\text{accelerating head}) = \frac{g}{l} h_a \tag{140}$$

The accelerating head as shown in equation (137) is $H - h - h_f$. It is the general principles of hydraulics that the head lost in flow through any opening, pipe, orifice, etc., varies as the square of velocity.

It was shown in Section 144, page 302, that the quantity flowing through a turbine varies as the square root of the head. Remembering that the quantity is proportional to the penstock velocity, we have:

$$\frac{q}{Q} = \frac{v}{V} = \frac{\sqrt{h}}{\sqrt{H'}} \text{ from which} \tag{141}$$

$$h = \frac{v^2}{V^2} H' \quad \text{Now} \tag{142}$$

$$h_f = (1 + f\frac{l}{d} + \text{etc.}) \frac{v^2}{2g} * \quad \text{Hence,} \tag{143}$$

$$\frac{h_f}{h_F} = \frac{v^2}{V^2} \quad \text{Or} \tag{144}$$

$$h_f = \frac{v^2}{V^2} h_F \tag{145}$$

* See Merriman's Treatise on Hydraulics, p. 200, et seq.

From equation (137)

$$h_a = H - h - h_f = H - H' \frac{v^2}{V^2} - h_F \frac{v^2}{V^2} \text{ or}$$

$$(146) \qquad h_a = H - (H' + h_F) \frac{v^2}{V^2}$$

And from equation (136)

$$(147) \qquad h_a = H - H \frac{v^2}{V^2} = H \left(1 - \frac{v^2}{V^2}\right)$$

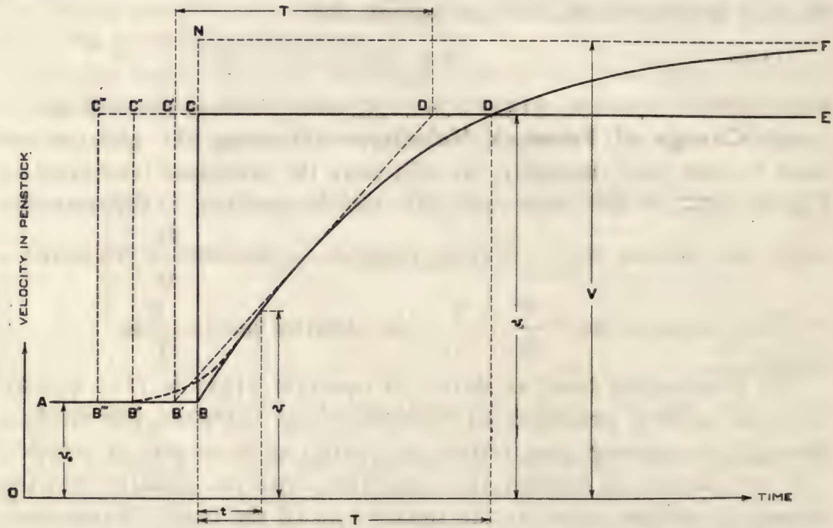

Fig. 278.—Form of Curve of Velocity Change for Increasing Velocity (see page 425).

Hence from equation (140)

$$(148) \qquad \frac{dv}{dt} = \frac{gH}{l} \left(1 - \frac{v^2}{V^2}\right)$$

The integration of this equation as given in Section 213 gives the following equation for the curve of velocity change in the penstock following a sudden change of gate opening:

$$(149) \qquad v = V \frac{B \text{ anti log } k't - 1}{B \text{ anti log } k't + 1}$$

As shown in Section 213 this value of v approaches but never equals the value of V. The form of the curve for an increasing velocity is shown in Fig. 278.

199. Effect of Slow Acceleration on Water Supplied to Wheel.—Since velocity in the penstock, discharge of wheel, and load are approximately proportional to each other, the ordinates of Fig. 278 may be taken to represent loads. The load demand remains at a constant value v_0 from *A* to *B*, where it suddenly increases to v_1 following the line *A B C D E*. The supply, however, assuming an instantaneous gate movement, follows the line *A B D F*. Now, the total quantity of water supplied to, and hence the work (not power) supplied by the water for all purposes is proportional to the area generated by an ordinate to the line *A B D F*, and the demand upon the wheel to the area generated by the power curve. The area *B C D B* therefore represents a deficiency of developed work which must be supplied by the energy stored in the rotating parts.

For practical purposes this area may be assumed equal to the area *L* of the triangle *B′ C′ D′*, where the line *B′ D′* is tangent to the curve *B M D* at the point of mean velocity $\frac{v_0 + v_1}{2}$

The slope of the line *B′ D′* for this mean velocity is readily obtained from equation (148). Call it *M*, then

$$\text{(150)} \qquad M = \frac{B'C'}{C'D'} = \frac{v_1 - v_0}{T'} = \frac{gH}{1}\left[1 - \frac{(v_0 + v_1)^2}{4V^2}\right] \text{ and}$$

$$\text{(151)} \qquad T' = \frac{v_1 - v_0}{M}$$

$$\text{(152)} \qquad \text{Area } B'C'D' = L = \frac{(v_1 - v_0)T'}{2} = \frac{(v_1 - v_0)^2}{2M}$$

This value of *L* is expressed in feet and represents the deficiency of lineal distance moved by the water column in the penstock. The deficiency of supplied water in cubic feet is, hence, *A L* and the deficiency of undeveloped work is

$$\text{(153)} \qquad \triangle K_1 = ALwH = \frac{AwH}{2M}(v_1 - v_0)^2$$

200. Value of Racing or Gate Over-Run.—At *D*, Fig. 278, page 424, the supply line *B D F* crosses the load line *C D E*, and the speed which was lost from *B* to *D* begins to pick up again.

The necessity also for an overrun of the governor is shown by Fig. 278. If the demand line were *A B N F* and the gate opened to the same place as before, giving the supply line *B D F*, the supply of power would approach, but theoretically never equal, the demand and the speed would hence never pick up to normal. The gate movement

should therefore be similar to that shown in Fig. 279 in order to give the gate the small overrun which is necessary to bring the speed back to normal.

201. Energy Required to Change the Penstock Velocity.—The energy involved in the change of velocity above described results in an excess or deficiency of energy delivered to the wheel (see Section 192). The amount of this excess or deficient energy is readily determinable. The kinetic energy in foot pounds stored in the moving column of water is $K_2 = \frac{Wv^2}{2g}$ or

$$K_2 = \frac{62.5Alv^2}{64.3} = .972\ Alv^2 \qquad (154)$$

The amount which must be diverted from the wheel or dissipated when the velocity changes is therefore

$$\triangle K_2 = 0.972\ Al\ (v_1^2 - v_0^2) \qquad (155)$$

In this case l should be taken as the combined length of penstock and draft tube.

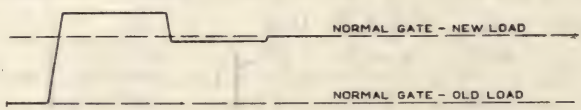

Fig. 279.—Gate Movement Showing Necessary Overrun.

This deficient energy must be supplied, or the excess absorbed, by means of a flywheel or the installation of a stand-pipe connected with the penstock closely adjoining the wheel.

202. Effect of Sensitiveness and Rapidity of Governor.—Referring again to Fig. 278, page 424, suppose the increase of load to take place at B''' giving the load line $A\ B'''\ C'''\ E$. After an interval from B''' to B'', the speed has dropped an amount depending upon the sensitiveness of the governor. The gate will then begin to open; the velocity in the penstock accelerating meanwhile along the dotted line $B'''Y$. The lack of sensitiveness of the governor has therefore added a deficient work area of $B'''\ B''\ C''\ C'''$, and the sluggishness of its motion an additional area $C''\ B''\ B\ C$, approximately. This deficiency $\triangle K_3$ can be only roughly approximated without the detailed analysis given in Section 213.

203. The Fly-Wheel.—A fly-wheel is valuable for the storage of energy. Work must be done upon it to increase its speed of rotation, and it will again give out this energy in being retarded. From the

laws of mechanics the number of foot pounds of kinetic energy stored in a body by virtue of its rotation is given by the formula:

$$K' = \frac{2 I \pi^2 S^2}{g\, 60^2} = \frac{2 \times 3.1416^2}{32.15 \times 60^2} I\, S^2 \text{ or}$$

(156) $$K' = .00017\, I\, S^2$$

The amount of energy which must be given to or absorbed from the fly-wheel in order to change the speed is

(157) $$\triangle K' = .00017\, I\, (S_0^2 - S_1^2)$$

Thus a fly-wheel can store energy only by means of a change in speed. By means of a sufficiently large moment of inertia the speed change of a fly-wheel, for any given energy storage $\triangle K'$, can be reduced to any desirable limit.

The need of a fly-wheel effect to carry the load of a hydro-electric unit during changes of gate, and while the water is accelerating in the penstock at an increase of load has led to the development of a type of revolving field generator, whose rotor has a high moment of inertia and is therefore especially adapted for speed regulation usually making the use of a fly-wheel unnecessary.

Warren * has simplified the expression for $\triangle K'$ (see equation 157), substantially as follows:

From equation (156):

(158) $$\frac{K_1'}{K_2'} = \frac{.00017\, I\, S_1^2}{.00017\, I\, S_2^2} = \frac{S_1^2}{S_2^2} \quad \text{Hence,}$$

(159) $$\frac{K_1' - K_2'}{K_2'} = \frac{S_1^2 - S_2^2}{S_2^2} = \frac{(S_1 + S_2)\,(S_1 - S_2)}{S_2^2}$$

$$\text{Put } S_1 - S_2 = \triangle S$$
$$\text{and } K_1' - K_2' = \triangle K'$$

For small differences between S_1 and S_2 equation (159) becomes approximately:

(160) $$\frac{\triangle K'}{K'} = \frac{2S \times \triangle S}{S^2} = \frac{2 \times \triangle S}{S} \text{ or}$$

(161) $$\triangle K' = \frac{2K' \times \triangle S}{S}$$

Or the percentage change in speed is

(162) $$\delta = \frac{100 \times \triangle S}{S} = \frac{50 \times \triangle K'}{K'}$$

* See "Speed Regulation of High Head Water Wheels," by H. E. Warren, in Technology Quarterly, Vol. XX, No. 2.

204. The Stand-Pipe.—The function of the stand-pipe is two-fold: (1) to act as a relief valve in case of excess pressures in the penstock; (2) to furnish a supply of energy to take care of sudden increases of load while the water is accelerating, and to dissipate the excess kinetic energy in the moving water column at time of sudden drop in load. For these purposes it should be of ample diameter and placed as close to the wheel as possible.

The analytical determination of the effect of a given stand-pipe upon speed regulation is very difficult if not quite impossible. Furthermore, it is not necessary, since the drop in effective head at an increase of load may (except in the case of maximum possible load) be compensated for by an increase of gate opening, hence maintaining a constant power and speed or at least a satisfactory degree of speed regulation. Thus the action of a stand-pipe in storing energy differs radically from that of the fly-wheel as the latter can store or give out energy only by means of a change of speed in the generating unit.

The determination of the range of fluctuation of water level in an assumed stand-pipe, and the time required for return to normal level for various changes of load on the wheel, will assist greatly in the design of the stand-pipe.

Figure 280, page 429, shows the condition when a stand-pipe is used. Assume that the wheel is operating under part load. The water normally stands at height h_F below the supply level. If the load suddenly increases, the gates open, and the water level begins to fall, thus causing an accelerating head $h_a = H - h - h_f$. Equation (140) then applies as before, where h_a becomes $(h - cv^2)$.

If the governor keeps step with the change in head by increasing the gate opening to maintain a constant power then

$$q\,h = q_1\,h_1$$

$$q\,(H - y) = Av_1\,(H - h_F) = Av_1\,(H - cv_1^2) \text{ or}$$

$$q = \frac{Av_1\,(H - cv_1^2)}{H - y} \tag{163}$$

The rate of water consumption by the wheel at any instant is q; the rate at which the water is supplied by the penstock is Av; and the rate of rise or fall of the water surface in stand-pipe is therefore:

$$\frac{dy}{dt} = \frac{dh}{dt} = \frac{Av - q}{F} = \frac{A}{F}\left[v - \frac{v_1\,(H - cv_1^2)}{H - y}\right] \tag{164}$$

The solution of equations (140) and (164), which are necessary for determining the curves of variation of head and velocity, is im-

practicable, if not impossible, hence a different treatment is proposed and considered in Section 214.

If q be assumed constant ($= Av_1$) during the adjustment of penstock velocity and the friction loss cv^2, in the penstock be neglected, then equations (140) and (164) simplify and become integrable. The resulting equations, showing the variations of v and y, are true harmonics or sine curves. The effect of friction and governor action is to produce a damped or somewhat distorted harmonic as discussed in Section 213. Any change of load thus starts a series of wave-like fluctuations of penstock velocity and stand-pipe level which continue

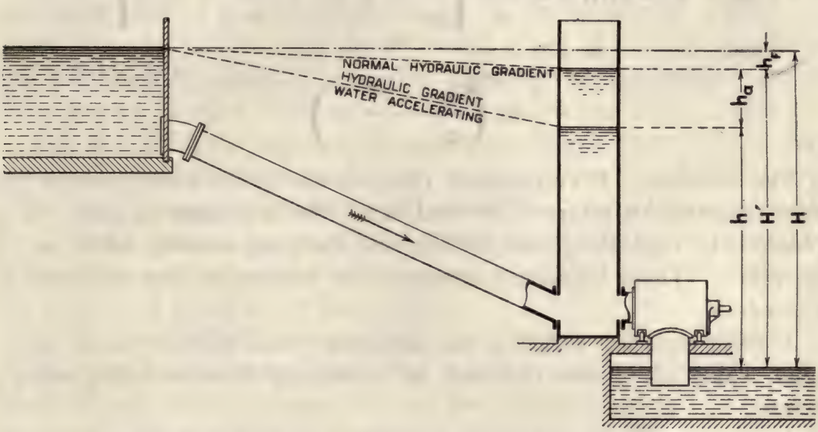

Fig. 280.—The action of the Stand-pipe (see page 428).

until this wave energy has been entirely expended in friction. Analogous to all other wave motions these waves may pile up (if two or more gate movements succeed each other by short intervals which are approximately multiples of the cycle $2T$), causing a very great fluctuation in head and velocity. In fact by assuming a proper combination and succession of circumstances no limit can be assigned to the range of fluctuation or "surge" which may occur. The probable combination of circumstances which will occur in any plant depends largely upon the character of the load. Overflows from stand-pipes due to these surges have been known to do considerable damage and it is desirable to either provide for this overflow either at the top or by relief valves at the bottom, or build the stand-pipe high enough to prevent it and thus gain the additional advantage of conserving the water which would otherwise waste.

If the change of load is assumed to occur when the water is at its normal level then the analysis given in Section 214 furnishes the following formulas:

$$T = \pi \sqrt{\frac{Fl}{Ag}} \tag{165}$$

$$Y = \sqrt{\frac{Al}{Fg}}\,(v_1 - v_0) \tag{166}$$

$$D^2 - 2HD = -\frac{2A}{F}\left[\frac{1}{2g}(v_1^2 - v_0^2) + \frac{HT}{\pi}(v_1 - v_0)\right] \tag{167}$$

$$D_b = D + c\,v_0^2 \tag{168}$$

$$D_a^2 = \frac{A}{F} v_0^2 \left(\frac{1}{g} - \frac{cT}{3} v_0\right) \tag{169}$$

The value of T from equation (165) is one-half a wave cycle or the time required for return to normal head after a change of load. It is obtained by neglecting both friction and the compensating effect of the governor. These influences increase T in very nearly the ratio that D exceeds Y.

Y from equation (166) is the maximum head fluctuation, or maximum value of y, also obtained by neglecting friction and governor action.

D from equation (167) is the maximum drop in stand-pipe level corresponding to Y except that governor action is included. If this value of D is added as shown in equation (168) to the initial friction loss cv_0^2, the result agrees very closely with the value of the maximum drop D, where friction is included and is much more simple than the more exact equation given in Section 214.

A reasonable assumption for determining the probable maximum height to which the water will rise in the stand-pipe is that full load is instantly thrown off the unit when the normal load velocity v_f, exists in the penstock. This assumption leads to equation (169).

The verification of these formulas and some additional ones is given in Section 214, and an example of their application in Section 211.

205. The Air Chamber.—There is a practical limit to the height to which a stand-pipe can be built. A high stand-pipe is also less effective due to the inertia of the water in the stand-pipe itself which must be overcome at each change of load, thus introducing to a lesser degree the same problem as in a penstock without stand-pipe. For

some such cases the top of the tank can be closed and furnished with air by a compressor. The design of air chambers has been investigated by Raymond D. Johnson.* An air chamber is less effective in equalizing the pressure than a stand-pipe of the same diameter.

206. Predetermination of Speed Regulation for Wheels Set in Open Penstocks.—The influences which oppose speed regulation have been partly discussed. At an increase or decrease of load there is a deficiency or excess of developed power due to (1) the inability of the governor to move the gate upon the instant that the load changes; (2) the necessity of accelerating or retarding the water in the penstock and draft tube as previously discussed. If no stand-pipe is used, reliance must be placed upon the fly-wheel effect of turbine, generator and additional fly-wheel, if necessary, to absorb or give out the excess or deficiency of input over output of the plant at this time.

The first influence opposed to speed regulation, that of slow gate movement, is of chief importance (a) where the plant is provided with large open penstocks and short draft tubes; (b) where an ample stand-pipe, placed close to the wheel, and a short draft tube are used; (c) in the regulation of an impulse wheel where no attempt is made to change the velocity of water in the feeder pipe.

Mr. H. E. Warren † has analyzed this case essentially as follows:

"As long as the output from the wheel is equal to the load, the speed S, and kinetic energy K', of the revolving parts will remain constant. The governor is designed to adjust the output of the wheel to correspond with the load, but it cannot do this instantaneously. Consequently, during the time T required to make the adjustment of the control mechanism after a load change there will be a production of energy by the water wheel greater or less than the load. The entire excess or deficiency will be added to or subtracted from the kinetic energy of the revolving parts, and will become manifest by a corresponding change in speed.

Neglecting friction losses, and assuming that the power of the water wheel is proportional to the percentage of the governor stroke and that the movement of the governor after a load change is at a uniform rate, the excess or deficient energy which goes to or comes from the revolving parts after an instantaneous change of load from L_0 to L_1 is measured by the average difference between the power of the wheel

* See Trans. of Am. Soc. M. E., 1908.

† See article by H. E. Warren on "Speed Regulation of High Head Water Wheels," previously referred to in Section 203.

and the new load during the time T'', while the governor is moving, multiplied by T'' or expressed in foot pounds:

$$(170) \qquad \triangle K' = \frac{p_0 - p_1}{2} \times T'' \times 550$$

From equation (156) the kinetic energy of the rotating parts is:

$$K' = .00017 \, IS^2$$

From equations (156), (162) and (170)

$$\delta = \frac{50 \times (p_0 - p_1) \, T'' \times 550}{2 \times .00017 \, I \, S^2} \text{ or}$$

$$(171) \qquad \delta = 81{,}000{,}000 \frac{T''}{I \, S^2} (p_0 - p_1)$$

207. Predetermination of Speed Regulation, Plant With Closed Penstock.—In this case the rotating parts must absorb or deliver up an amount of energy $\triangle K'$ (equation 161), equivalent to that given for $\triangle K$ in formula

$$(139) \qquad \triangle K = \triangle K_1 + \triangle K_2 + \triangle K_3$$

where, from equation (153),

$$(153) \qquad \triangle K_1 = \frac{AwH}{2 \, M} (v_1 - v_0)^2$$

M being obtained from equation

$$(150) \qquad M = \frac{gH}{l} \left[1 - \frac{(v_0 + v_1)^2}{4V^2} \right]$$

The value of $\triangle K_2$ is obtained by equation

$$(155) \qquad \triangle K_2 = 0.972 \, Al \, (v_1^2 - v_0^2)$$

There is no simple way, as discussed in Section 202, of determining K_3. It must be estimated or analyzed graphically as in Section 214.

From equation

$$(156) \qquad K' = .00017 \, I \, S^2$$

If R is the proportion of this theoretical energy which is given to the rotating parts at a decrease in load, or which the rotating parts must give out during an increase of velocity and load, then

$$(172) \qquad \triangle K' = R \times \triangle K$$

and we have from equation

$$(162) \qquad \triangle K' = \frac{50 \times R \times \triangle K}{K'}$$

$$= \frac{50 \times R \times \triangle K}{.00017 \, I \, S^2} \text{ or}$$

$$\delta = 294{,}000 \frac{R \times \triangle K}{I\, S^2} \tag{173}$$

Solving for I we find the moment of inertia of the rotating parts, which is necessary to obtain any desired percentage of regulation to be

$$I = 294{,}000 \frac{R \times \triangle K}{\delta\, S^2} \tag{174}$$

Although there can be no doubt as to the accuracy of the form of equations (173) and (174) yet their value for other than comparative purposes depends upon the accuracy with which we can estimate R. With perfect efficiency of the wheel under all conditions, R would be unity, but in actual cases R must be determined by experiment or by the graphical method given in Section 213. It will be less for decreasing than for increasing loads since the inefficient operation of the wheel assists speed regulation in the former case, and hinders it in the latter. In addition to this fact, the excess energy at a decrease of load can be partially dissipated through a relief valve, or a by-pass, etc. For practical cases it is therefore necessary to investigate only the case of increasing load.

A detailed analysis of a particular problem can be made, as in Section 213, by which the velocity in the penstock, effective head, power of wheel, speed, etc., can be determined for each instant during the period of adjustment. From this also the time of return to normal speed can be determined. The method is somewhat tedious, but justifiable nevertheless.

208. Predetermination of Speed Regulation, Plant With Stand-pipe.—If the stand-pipe is of suitable diameter and close to the wheel the speed regulation will approach that obtainable in open penstock and as investigated by Warren in Section 206. Otherwise the problem becomes that of a plant with a closed penstock, of a length equal to that of the draft tube, plus the penstock from stand-pipe to wheel.

209. Application of Method, Closed Penstock.—An example of the analysis of a problem in speed regulation is as follows:

Assume the forty-eight inch Victor cylinder gate turbine, whose characteristic curve is shown in Fig. 281, page 434. Suppose it is supplied with water through a penstock whose diameter is eight feet, and whose length combined with that of the draft tube is 500 feet. The head is fifty feet which for $\phi = .664$ gives 180 R. P. M. $= S$.

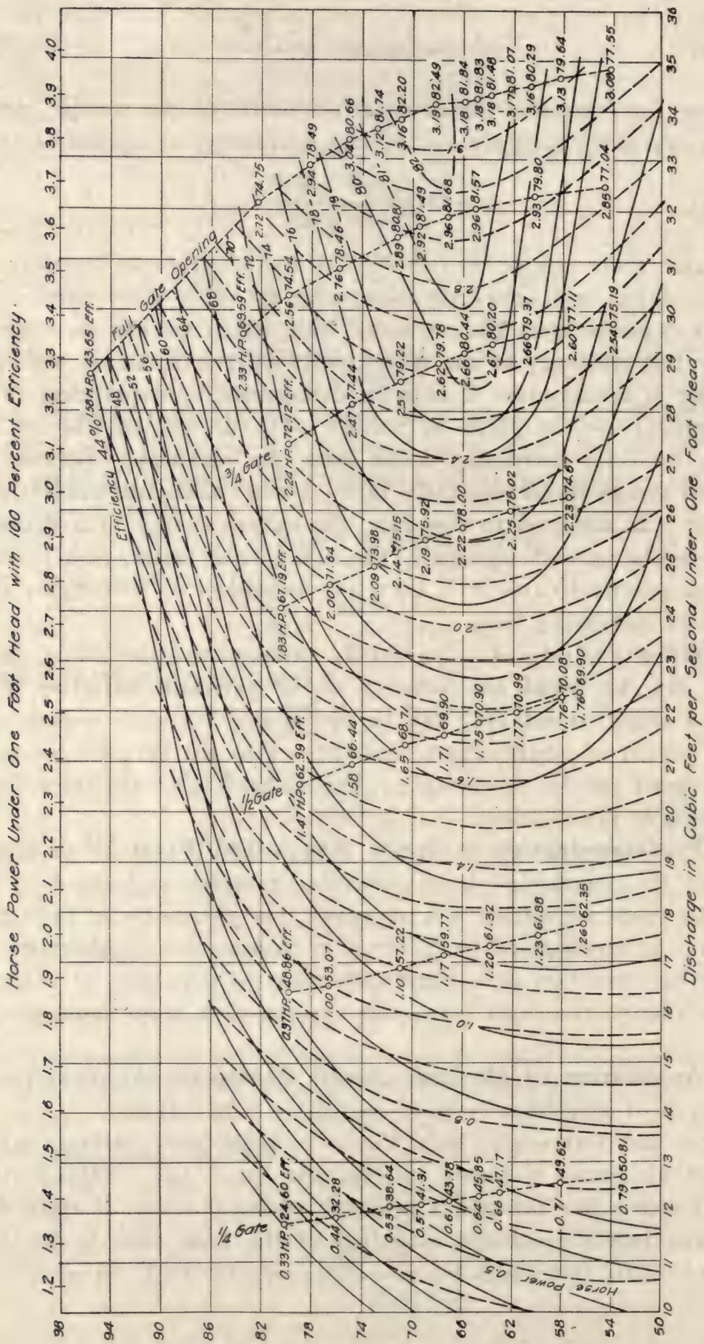

Fig. 281.—Characteristic Curve of 48-inch Victor Turbine With Cylinder Gate (see page 433).

Neglecting all losses of head except that in the turbine, we find from the characteristic curve for various loads as follows:

	Full load	.8 Load	½ Load	¼ Load
Brake horse power	1120.00	900	560.00	280.00
Quantity of water per sec. (cu. ft.)	240.00	210	145.00	97.80
Velocity in penstock, V	4.77	4.18	2.88	1.94
Efficiency of wheel	.82	.754	.68	.505

The above values will be considered as applying to the entire plant since the loss in the penstock is small in this case.

Assume the load to increase suddenly from one quarter load to eight-tenths load, while the gate at the same time opens to full load position. The number of foot pounds of work which must be done to accelerate the water from a velocity of 1.94 feet per second to 4.18 feet per second is found from equation (155) to be

$$\begin{aligned}\triangle K_2 &= 0.972 \text{ Al } (v_1^2 - v_0^2)\\ &= 0.972 \times 50.3 \times 500 \ (4.18^2 - 1.94^2)\\ &= 0.972 \times 50.3 \times 500 \times 13.73\\ &= 335{,}000 \text{ foot pounds.}\end{aligned}$$

Referring to Section 207, page 432, to find the amount of deficient work due to insufficient supply of water we have

$$\frac{v_0 + v_1}{2} = 3.06,$$

From equation (150), Section 199

$$\begin{aligned}M &= \frac{32.15 \times 50}{500}\left(1 - \frac{3.06^2}{4 \times 4.77^2}\right)\\ &= \frac{32.15 \times 50}{500} \times .897\\ &= 2.88\end{aligned}$$

From equation (153),

$$\begin{aligned}\triangle K_1 &= \frac{50.3 \times 62.5 \times 50}{2 \times 2.88}(4.18 - 1.94)^2\\ &= 187{,}000 \text{ foot pounds.}\end{aligned}$$

The total deficiency for which formulas have been derived is hence,

$$\begin{aligned}\triangle K &= \triangle K_1 + \triangle K^2 + (\triangle K_3 \text{ undeterminable})\\ &= 335{,}000 + 137{,}000\\ &= 472{,}000 + \text{ ft. lbs.}\end{aligned}$$

By means of the detailed graphical analysis given in Section 213 this deficiency is found to be 600,000 foot pounds for gate movement in one-half second showing that the estimated value should have been increased in this case by 12.7 per cent. ($R = 1.127$) to compensate for neglecting the effect of slow (one-half second) gate movement, or K_3. It must be remembered that this quantity $\triangle K$, is the deficiency of theoretical hydraulic work done upon the wheel. For reasons discussed in Section 213, it will, however, be found to differ but slightly from the deficiency of wheel output, in this case 586,000 foot pounds.

To determine the speed regulation which can be obtained, assume a generating unit whose rotor has a fly-wheel effect, or moment of inertia I, of 1,000,000 pounds at one foot radius. The normal speed $S = 180$, $\triangle K = 472{,}000$ foot pounds, and R (in general to be estimated, but in this case obtained by the graphical method given in Section 213, is 1,127. Therefore from equation (173)

$$6 = 294{,}000 \frac{1.127 \times 472{,}000}{1{,}000{,}000 \times 180^2} = 5.42\%$$

If a fly-wheel is to be designed for a given regulation, say four per cent., then the required moment of inertia of same is, from equation (174).

$$I = 294{,}000 \frac{R \triangle K}{S^2}$$

$$= 294{,}000 \frac{1.157 \times 472{,}000}{4 \times 180^2} \text{ or}$$

$$I = 1{,}355{,}000 \text{ ft.}^2 \text{ lbs.}$$

210. **Application of Method, Open Penstock.**—As the penstock and draft tube are shortened, the excess or deficient energy area $\triangle K_3$, obtained during the gate movement becomes an increasing proportion of the whole, until for a large open penstock and short draft tube the developed power ceases to lag and follows practically the same law of change as the gate opening. The estimation of excess or deficient energy, and consequently of speed, is then very simple by means of Mr. Warren's equation (171). For illustration: assume the same wheel as in the preceding section, obtaining the outputs of 280 H. P. $= P_0$ at one-fourth load and 1,120 H. P. $= P_1$ at full load, as in the other installation. Assume the same moment of inertia 1,000,000 and that the gate movement takes place in one-half second as before. Then $T'' = \frac{1}{2}$; $S = 180$.

This gives

$$\delta = 81{,}000{,}000 \frac{0.5}{1{,}000{,}000 \times 180^2} (1{,}120 - 280) = 1.05\%$$

This is a much closer regulation than obtained with the long penstock.

211. Application of Method, Plant With Stand-Pipe.—Assume a plant where the wheels develop 39,000 H. P. under 375 ft. head, thereby requiring about 1,100 cubic feet of water per second (assuming eighty-three per cent. efficiency of the wheels). Assume this water is supplied through four seven foot pipes about 4,800 feet long, requiring a velocity in the feeder pipes at full load of about 7.15 feet. Suppose four pipes all connected at the lower end to a stand-pipe thirty feet in diameter. If a sudden load change, of about one-third of the total is to be provided for, this would require an ultimate change of velocity in the penstock from about 4.76 feet per second at two-thirds load to 7.15 feet at full load, or $v_0 = 4.76$, and $v_1 = 7.15$. Now,

$$A = 4 \times \pi \frac{7^2}{4} = 154 \text{ sq. ft.}$$

$$F = \pi \frac{30^2}{4} = 707$$

From equation (165) the time required for return to normal head, or the half period of oscillation, is

$$T = \pi \sqrt{\frac{707 \times 4800}{154 \times 32.15}} = 82 \text{ seconds}$$

This would perhaps be increased to nearly 100 seconds, due to the use of additional water during this period of low head, as discussed in Section 214, but the value eighty-two should be used in equation (167).

Equation (166) gives for the drop in water level in the stand-pipe,

$$Y = \sqrt{\frac{154 \times 4800}{707 \times 32.15}} (7.15 - 4.76)$$

$$= \sqrt{32.5} \times 2.39 = 13.6 \text{ feet.}$$

The more exact equations (167) and (168), give for D and D_b

$$D^2 - 2 \times 375\, D = -\frac{2 \times 154}{707} \left[\frac{4800}{64.3} (7.15^2 - 4.76^2) + \frac{375 \times 82}{3.14} (7.15 - 4.76) \right]$$

or

$$D^2 - 750\, D + 11{,}120 = 0$$

Solving this quadratic equation gives

$$D = \frac{750 - \sqrt{750^2 - 4 \times 11{,}120}}{2} \text{ or}$$

$$D = \frac{750 - 719}{2} = 15.5 \text{ feet}$$

$$C = \left(1 + .015 \frac{4800}{7}\right) \frac{1}{64.3} = .176$$

$$D_b = 15.5 + c \times 4.76^2 = 15.5 + .176 \times 4.76^2 = 19.5 \text{ feet}$$

No attempt will be made to estimate the greatest drop in level which might occur, due to an addition of waves.

212. Further Consideration of Water Hammer.—In Section 194, it is shown that the pressure head due to a change of velocity in a water column is expressed by the formula

$$h_a = \frac{1}{g} \times \frac{dv}{dt} \tag{138}$$

It is evident that the water hammer head produced by the rapid closing of a gate at the end of a pipe line will be maximum for the maximum possible value of $\frac{dv}{dt}$, or that obtained by closing the gate instantly. Were it not for the elasticity of water and pipe, instantaneous gate closure would produce an infinite rate of retardation, $\frac{dv}{dt}$, and hence infinite pressure. In reality the water near the gate first compresses and the surrounding pipe expands, due to the water hammer pressure, the flow meanwhile continuing undiminished in the remainder of the pipe in order to fill the additional space thus obtained. The point up to which this compression of the water has taken place, as shown by Joukowsky * travels along the pipe from gate to reservoir as a wave with a velocity λ,† equal to that of sound

$$\lambda = \frac{12}{\sqrt{\frac{w}{g}\left(\frac{1}{K} + \frac{d}{eE}\right)}}$$

* See the "Memoirs of the Imperial Academy of Sciences of St. Petersburg," vol. IX, No. 5. Ueber den Hydraulischen Stoss in Wasserleitungsröhren, by N. Joukowsky; published in German and Russian. See also the synopsis of same by O. Simin in The Trans. of the American W. W. Ass'n, 1904.

† λ varies from about 4,500 to 3,000 feet per second as the size of the pipe increases, and can always be obtained by the formula (due to Joukowsky):

where:

λ = velocity of the wave in feet per second.

K = volumnar modulus of elasticity of the water = 294,000 pounds per square inch.

e = thickness of the pipe walls in inches.

E = modulus of elasticity of the material of the pipe.

w, g, and d = as previously defined.

in the same column of water. The water has not all been brought to rest until the wave reaches the reservoir, which evidently requires a time $\frac{l}{\lambda}$. Although only an elementary length of the water column is brought to rest at a time, the effect upon the pressure is the same as would result from retarding the whole column as a unit in a time $\frac{l}{\lambda}$. The maximum possible rate of retardation is hence

$$\text{(175)} \qquad \text{Max.} \frac{dv}{dt} = v \div \frac{l}{\lambda} = \frac{v\lambda}{l}$$

From equation (138)

$$\text{(176)} \qquad H_m = \text{maximum } h_a = \frac{l}{g} \cdot \frac{v\lambda}{l} = \frac{\lambda v}{g} \text{*}$$

The pressure head given by this formula varies from about 140 to 100 feet per foot of extinguished velocity as the pipe increases in size from two inches upwards. If the gate is only partially closed by this instantaneous motion, the pressure head is given by the same formula in which case *v* represents the amount of the velocity which is instantaneously extinguished.

Thus, in the case of instantaneous gate movement, the pressure is not produced at the same instant along the entire pipe, but travels as a wave with a velocity λ from the gate to the origin of the pipe and back again to the gate. It then reverses and becomes a wave of rarefaction which travels twice the length of the pipe in the same manner. This continues until the energy of the moving column of water has been dissipated by friction, and the wave gradually subsides. This phenomenon is identical with that of the vibrating sound wave in an organ pipe.

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* This formula is the same as that obtained by Joukowsky by two other methods of analysis. His discussion of water hammer phenomena includes all that is known upon the subject and it, or Simin's synopsis, should be read especially by every engineer interested in high head developments as the subject can only briefly be touched in this book.

Although equation (176) gives the maximum possible pressure head which can result from the extinction of a given velocity v, in a pipe it does not, however, represent the maximum pressure which could be obtained as the result of several successive gate movements; in fact, no limit can be assigned to the pressure which might result in case several water hammer waves were to be produced at intervals differing approximately by multiples of the vibration period of the water column, in which case they are known to "pile up" to enormous indeterminable pressures.

When the flow in a pipe is shut off by the gradual closure of a gate then equation (138) and also the following equation

$$\text{(177)} \qquad \frac{dv}{dt} = \frac{gH}{l}\left(1 - \frac{v^2}{V^2}\right)$$

from Sections 194 and 198, apply as before except that in this case not only v but also V is a variable, its value being different for each successive position of the gate, and its law of variation depending upon the law and rate of gate movement. The integration of equation (177) in its general form, to obtain the velocity curve is then very difficult if not impossible.

An approximate curve of v, and hence also of h can be plotted by assuming the gate closure to take place by means of a great many small instantaneous movements, according to any law which may be chosen. The value of V for each of the many gate positions can then be computed from the known hydraulic data of the wheels and penstock.

Now, in equation (177), substitute for v the initial velocity in the pipe, and for V the normal velocity (above determined), after the gate has received its first small instantaneous movement. The result will be the initial slope of the v–t curve $= \frac{dv}{dt_0}$. Assume this rate of decrease in velocity to continue constant for the short interval between successive gate movements; then the actual velocity v, at the instant of the next gate movement will be

$$\text{(178)} \qquad v = v_0 - i\frac{dv}{dt_0}$$

where i is the interval between the two movements.

Assume this new value of v, to be v_0 and using the value of V for the corresponding (or second) gate position, again apply equations (177) and (178), until the gate is completely shut.

Having thus determined the v–t curve, the head curve can be readily found from equation (138), which gives the excess of head above static or so called water hammer head.

Substituting the value of $\frac{dv}{dt}$ from (177) in (138) give

$$h_a = H\left(1 - \frac{v^2}{V^2}\right) \qquad (179)$$

Church has investigated this problem by a method described in the Journal of the Franklin Institute for April and May, 1890.

213. A More Detailed Analysis of Speed Regulation.—In Section 198, the following equation was shown to express the rate of acceleration of water in the penstock subsequent to an instantaneous change in gate opening of the wheel.

$$\frac{dv}{dt} = \frac{gH}{l}\left(1 - \frac{v^2}{V^2}\right) \qquad (177)$$

Separating the variables v and t, gives

$$dt = \frac{lV^2}{gH} \cdot \frac{dv}{V^2 - v^2}$$

Integrating we have:

$$t = \frac{lV}{2gH} \log_e \frac{V - v}{V + v} + C \qquad (178)$$

To determine the constant of integration C, assume that $v = v_0$ when $t = 0$, hence

$$C = -\frac{lV}{2gH} \log_e \frac{V - v_0}{V + v_0}$$

Let

$$k = \frac{2gH}{Vl} \text{ and } k' = \frac{2gH}{2.3\ Vl} \qquad (179)$$

$$B = \frac{V + v_0}{V - v_0} \qquad (180)$$

Substituting these values of C, B and k in (178), gives,

$$t = \frac{1}{k} \log_e \frac{1}{B} \cdot \frac{V - v}{V + v} \qquad (181)$$

From the definition of a logarithm: if $X = log_e\ N$, then $e^x = N$ hence

$$e^{kt} = \frac{1}{B} \cdot \frac{V - v}{V + v} \qquad (182)$$

Solving for v we obtain:

$$v = V\frac{Be^{kt} - 1}{Be^{kt} + 1} \tag{183}$$

From the principle of logarithms we have:

$$e^{kt} = 10^{\frac{kt}{2.3}} = 10^{k't}$$

hence

$$v = V \cdot \frac{B \times \text{antilog } k't - 1}{B \times \text{antilog } k't + 1} \tag{184}$$

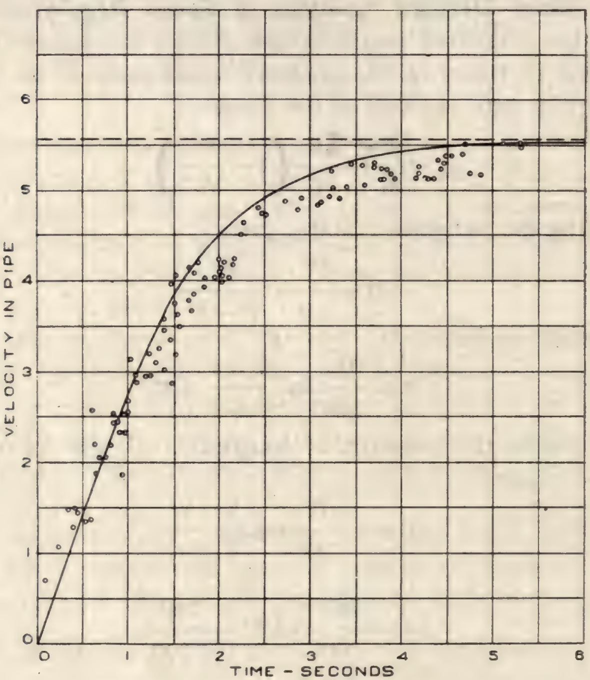

Fig. 282.—Curve Showing the Acceleration of Water in a Pipe Line After a Sudden Opening of the Gate.

Equation (184) is very readily applied to finding the curve of velocity increase or decrease in any pipe line subsequent to a sudden change of gate opening. It has been experimentally demonstrated for the acceleration of water in the drive pipe of an hydraulic ram, as shown by Fig. 282 which is taken from Bulletin No. 205, University of Wisconsin, Engineering Series, Vol. 4, No. 3, "An Investigation of the Hydraulic Ram" by L. F. Harza.

The curve is the plot of equation (184) and the experimental points were determined by an especially designed instrument. The fact that they fall commonly below the theoretical curve is due to a systematic friction error in the instrument. The agreement is sufficiently close, however, to entirely verify the form of equation (184).

Figure 283, page 445, shows the curves determined from equation (184) for the wheel used for illustrative problems in Section 209. Acceleration curves are shown for changes from 0 to the velocities of one-fourth, one-half, nine-tenths and full load; retardation curves from an initial velocity of five feet per second to the above velocities. It will be observed that in each case the actual velocity approaches, but theoretically never equals, the normal value V, for the given gate position.

The values of the constants used in computing these v–t curves are given below. B, for the accelerating from an initial velocity of zero, is:

$$B = \frac{V + v_0}{V - v_0} = \frac{V}{V} = 1$$

The other constants are: $H = 50'$, $l = 500'$, and $v_0 = 5'$ for retardation curves; also for the retardation curves B is negative, since v_0 is greater than V. If we always use the positive value of

$$B = \frac{V + v_0}{V - v_0}$$

we will obtain two equations:

For increasing velocities or *acceleration*

$$v = V\frac{\text{antilog } k't - 1}{\text{antilog } k't + 1} \tag{185}$$

For decreasing or *retarding* velocities,

$$v = V\frac{B \text{ antilog } k't + 1}{B \text{ antilog } k't - 1} \tag{186}$$

From equations (179) and (180) we obtain the table,

Load.	V.	B	k′
1.0	4.77	41.3	.585
.9	4.49	19.1	.623
.5	2.88	3.71	.975
.25	1.94	2.27	1.444

The computations of v, by equations (185) and (186), for various assumed values of t is very simple if tabulated as below. The com-

putation of the curve of acceleration and retardation of water in the penstock from 0, and from five feet per second, respectively, to its value 2.88 feet per second for one-half load is shown. It is assumed that the gate opens instantly from 0 to its position at one-half load, and closes to this position instantly when the velocity is five feet per second, giving the values of velocity in columns v and v', (4) and (6), respectively.

*Computation of v–t Curve.**

H = 50′, l = 500′, d = 8′, k′ = .975, B = 3.71, v_0 = 0 and 5′, V = 2.88′

(1)	(2)	(3)	(4) = v	(5)	(6) = v′
t	k′t	antilog of k′t	$\frac{(3)-1}{(3)+1}$ 2.88	(3) × 3.71	$\frac{(5)+1}{(5)-1}$ 2.88
.0	.0	1.	.0	3.71	5.0
.1	.0973	1.251	.321	4.64	4.17
.2	.1946	1.565	.635	5.81	4.077
.4	.3892	2.45	1.210	9.10	3.59
.6	.5838	3.835	1.690	14.23	3.31
.8	.7784	6.003	2.055	22.27	3.15
1.0	.973	9.397	2.327	34.85	3.05
1.2	1.168	14.72	2.513	54.70	2.99
1.4	1.362	23.01	2.64	85.5	2.95
1.7	1.654	45.08	2.753	167.3	2.91
2.0	1.946	88.31	2.81	328.0	2.897

Referring again to Fig. 283, page 445, we see that the acceleration curves thus computed all have a common tangent at the origin showing an initial rate of acceleration in each case of,

$$\frac{dv}{dt} = \frac{gH}{l}$$

The initial rate of retardation, however, depends upon the gate opening.

As shown by equations (185), (186), and the curves in Fig. 283, page 445, the velocity never equals but approaches indefinitely near, to its normal value *V*, for a given gate opening.

To show the application of the foregoing discussion to the change of penstock velocity, power, speed, etc., at a change of load, refer to Fig. 284, page 446. Here the line *AB* represents one-fourth load, line *C C* represents full load, line *D D* eight-tenths load and line *H H* forty-five per cent. load for the same wheel discussed above. Lines *A′ B′*, *C′ C′* and *D′ D′* represent the corresponding hydraulic power

* A number enclosed in parenthesis refers to the value given in the column of that number.

input lines. Line *abccba* represents the line of gate movement from its initial position at one-fourth to its position at full load and back again to one-fourth load. Line $O\,C_v\,C$ is copied from Fig. 283 and represents the curve of velocity increase which would result from a sudden complete opening of the gate. At b the gate begins to open, and the velocity to increase along an estimated curve $B_v\,C_v$. This curve could be more accurately determined by the process outlined in Section 212, but was not so determined here. In the same way curve

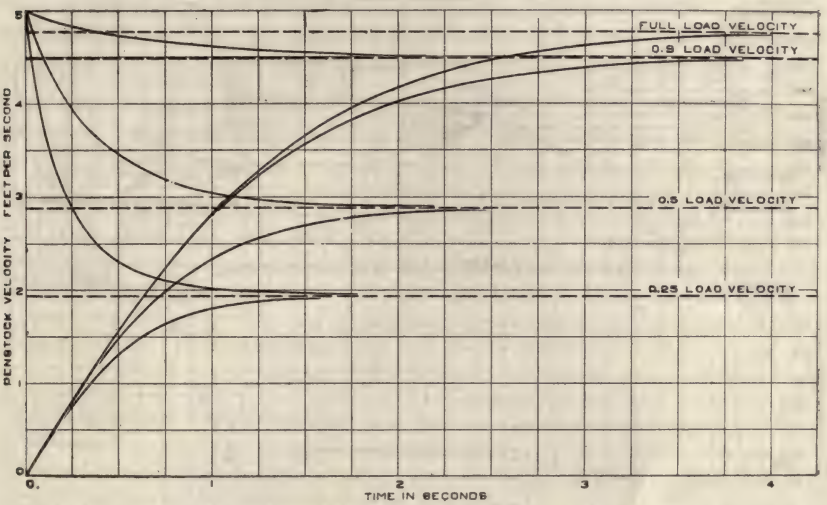

Fig. 283.—Curves of Acceleration and Retardation of Water in Penstock for Various Gate Movements (see page 443).

$F\,B'_v\,A_v$ was taken from Fig. 283 and the velocity curve during gate movement, $C'_v\,B''_v$ was estimated.

Having thus obtained the velocity curve $A_v\,B_v\,C_v\,C\,C'_v\,B'_v\,A_v$, the curve of effective head at the wheel can be readily determined from equation (142), Section 198, or

$$\text{(142)} \qquad h = \frac{v^2}{V^2} H'$$

While the gate is in motion from b to c the value of V changes, but can be readily estimated by interpolation from the values at one-fourth and full gates. From c to c (gate curve) V is constant, and equal to 4.77 feet per second. Since the friction loss in the penstock is slight in the problem under discussion H' is assumed to equal $H = 50'$. The resulting curve for h is $A_h\,B_h\,C_h\,C_h\,C'_h\,B'_h\,A_h$.

The curve of hydraulic horse power or *input* was then determined by applying the equation below to several points along the v and h curves obtaining curve $A' B' C_1 Y' X'$

$$P_1 = \frac{9h}{8.8} = \frac{Avh}{8.8}$$

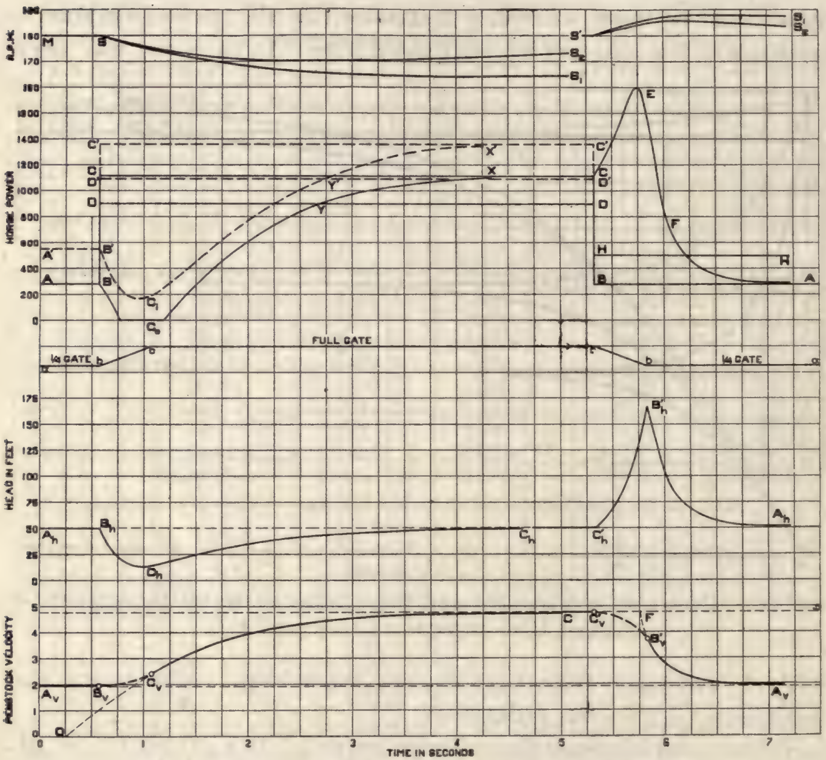

Fig. 284.—Graphical Analysis of Speed Regulation.

The output power curve $A\ B\ C_0\ Y\ X$ was then computed by

$$P = \frac{qhE}{8.8}$$

E or efficiency for each point was obtained from the characteristic curve of the wheel Fig. 281, page 434, by first computing from the known values of q, h, and S (= 180) at each point the values of the discharge under one foot head and ϕ.

Many interesting facts can now be seen from a study of Fig. 284. It will be seen that the opening or closing of the gate in order to in-

crease, or decrease, the power of the wheel has an immediate effect directly opposite to that intended and that in the output curve the power reduces to practically, if not quite, zero for nearly one-half second. The effective head drops very greatly during acceleration, and rises during retardation. It is evident that the rate of gate movement here used (one-half second) is too fast for closure, since the head rises to about 165 feet, over three times its normal value.

Now, since the product of power and time gives energy or work, it is evident that the areas of the figures generated by the ordinates to the various load curves are proportional to the demand for energy and the areas of the output curves are proportional to the supply. The area between the two curves, therefore, represents a deficiency or excess of work accomplished by the wheel, and can be measured by means of a planimeter or otherwise. The value of one square is $\frac{1}{4} \times 200 = 50$ H. P. seconds $= 50 \times 550 = 27{,}500$ foot pounds.

It was found in this way that the deficient hydraulic energy supplied to the wheel, assuming the load demand to increase from one-fourth to full is

$$27{,}500 \times \text{area } B' C_1 Y' X' C' B'$$
$$= 27{,}500 \times 36$$
$$= 990{,}000 \text{ foot pounds.}$$

The deficient load output is

$$27{,}500 \times \text{area } B C_0 Y X C B$$
$$= 27{,}500 \times 35 = 963{,}000 \text{ foot pounds.}$$

This deficiency of input over output must be supplied from the energy stored in the rotating parts, or from the fly-wheel effect, and can be accomplished only by a drop in speed of the power unit. Furthermore, in the case considered, the speed can never return to normal as long as the load remains at full value, but suffers a permanent drop due to the fact that v, q, h and power theoretically approach, but never equal the normal values for the new gate opening.

The excess energy, when the load again drops to its one-fourth value is,

$$27{,}500 \times \text{area } C E F A B C \text{ or}$$
$$27{,}500 \times 18 = 495{,}000 \text{ foot pounds.}$$

It is evident that this excess energy at decreasing load will always be less than the deficient energy at the time of increasing load, since the low efficiency of the wheel during the velocity-change tends to decrease the former and increase the latter.

It is also possible to dissipate the excess energy through a by-pass or relief valve, while no method is available for supplying the de-

ficiency during load increase except at a sacrifice of the kinetic energy of the rotating parts and consequent reduction of speed.

In Section 207 it was shown that the percentage departure of the speed from normal is

$$\delta = 294{,}000 \frac{R \times \triangle K}{I\,S^2}$$

Since the deficient energy $\triangle K$, is actually measured in this case, the estimated coefficient R, becomes unity. The normal speed S, of the wheel is 180, and I will be assumed as 1,000,000 square foot pounds, or 1,000,000 pounds at one foot radius, then

$$\delta = 294{,}000 \frac{963{,}000}{1{,}000{,}000 \times 180^2}$$
$$= 8.7 \text{ per cent.}$$

This is a permanent drop in speed.

In order for the speed to pick up again to normal, the gate must therefore overrun. The condition then is best illustrated by assuming in Fig. 284, page 446, that the load increases only to eight-tenths of full load value, following the line $A\,B\,D\,D$, while the gate movement follows the same line as before. In this case the v, h, wheel input, and wheel output curves will be unchanged.

The deficiency of input or of energy in the delivered water is then (by means of planimeter) represented by area $B'\,D'\,Y'\,C_1\,B'$ or

$$= 27{,}500 \times 21.8 = 600{,}000 \text{ foot pounds.}$$

The deficiency of output, represented by area $B\,D\,Y\,C_0\,B$, is

$$27{,}500 \times 21.3 = 586{,}000 \text{ foot pounds,}$$

giving a speed regulation of

$$= 294{,}000 \frac{586{,}000}{1{,}000{,}000 \times 180^2} = 5.32 \text{ per cent.}$$

The two quantities will probably always agree as closely as the accuracy of the problem demands, and much labor can be saved in an analysis if hydraulic horse power, or input, only is considered.

At Y the power curve crosses the demand line $D\,D$, and the speed begins to pick up, due to an excess of developed power. The time required for return to normal can be obtained by continuing the two curves until the excess area equals the former deficiency. In this case eight and one-half seconds is required.

By the successive application of equation (173) to narrow vertical strips of the excess or deficient energy area, we may plat the speed curve of the unit. In this way curve MSS_1, Fig. 284, for increase

from one-fourth to full load; curve MSS_2 for increase from one-fourth to eight-tenths load but simultaneous full gate opening; curve $S' S_1$, for decrease from full to one-fourth load, and curve $S' S_2$ for decrease from full to forty-five per cent. load, were platted. Curves MSS_1 and $S' S_1$ never returned to normal (180 R. P. M.), but curve MSS_2 returns in eight and one-half seconds, and curve $S' S_2$ in four seconds.

It is the belief of the writer that this method of analysis is not too long for a problem in practice and, if not, is therefore better than the method previously given since the conditions before and during gate movement can be readily included.

214. Further Consideration of the Stand-Pipe.—It was shown in Section 204, that the following equations apply to the operation of a plant with stand-pipe:

$$\frac{dv}{dt} = \frac{g}{l} \text{ (accelerating head) } = \frac{g}{l} h_a \tag{140}$$

$$\frac{dy}{dt} = \frac{dh}{dt} = \frac{Av - q}{F} \tag{164}$$

The value of h_a in a plant with penstock, is

$$h_a = y - h_f$$

Hence

$$= y - \left(1 + f\frac{l}{d} + \text{etc.}\right)\frac{v^2}{2g} = y - cv^2$$

$$\frac{dv}{dt} = \frac{g}{l}(y - cv^2) \tag{187}$$

Equation (164) gives the instantaneous rate of fluctuations of water level in the stand-pipe.

Equation (187) gives the rate of increase of penstock velocity in terms of the then existing values of y and v.

The quantity q, in equation (164), represents the water used by the wheel. This may remain practically constant if the head fluctuation is not too large, in which case the speed of the wheel will suffer; or, by means of an ideal action of the governor, it may be made to fluctuate inversely as the head h, thus maintaining a constant value of the product qh, and hence of the power input of the wheel. In case this latter assumption is made, then:

$$q\,h = q_1 h_1$$

or

$$q\,(H - y) = Av_1\,(H - cv_1^2)$$

Substituting this value of q in equation (164) gives:

$$\frac{dy}{dt} = \frac{A}{F}\left[v - \frac{v_1\,(H - cv_1^2)}{(H - y)}\right] \tag{188}$$

The solution of the two simultaneous differential equations (164) and (187), or (187) and (188), depending upon which assumption is made, is necessary in order to determine the exact curve of variation of head and velocity. Their general solution is however, very difficult if not impossible in this form. The equations may be applied successively to short portions of the arc by considering the curves to consist of a great many short straight lines. This method is not too long for application to a problem in practice, and will assist in obtaining approximate formulas which will be seen to coincide very closely with the true curves.

Assume an installation where $d = 8'$, $l = 500'$, $H = 50$, $F = 8A$. Let the velocities on the penstock at fractional loads be the same as given in the problem considered in Section 209. If the load suddenly increases from one-fourth to full, the velocity in the penstock must accelerate from 1.94 to 4.77 feet per second, or q from 97.8 to 240 cubic feet per second.

Estimating $f = .018$, equation (187) gives

$$\frac{dv}{dt} = \frac{32.15}{500}\left[y - (1 + .018\frac{500}{8})\frac{v^2}{64.3}\right] \text{ or}$$

$$\frac{dv}{dt} = .0643\,(y - .0331\,v^2) \tag{189}$$

and equation (188) gives:

$$\frac{dy}{dt} = \frac{dh}{dt} = \frac{v}{8} - \frac{4.77 \times 49.25}{8\,(H - y)} \text{ or}$$

$$\frac{dy}{dt} = \frac{v}{8} - \frac{29.4}{H - y} \tag{190}$$

Curves A_v and A_h, Fig. 285, page 453, show the curves of velocity v, and head h, respectively, obtained by applying equations (189) and (190) alternating to the two curves, considering them to remain straight for the time interval between consecutive points which were taken from one-fourth to one second apart depending upon the curvature. The closer these points are taken the more accurate would be the resulting curves.

If friction in the penstock, and the action of the governor, in compensating for the fluctuations of h, be neglected then equations (140) and (164) become

$$\frac{dv}{dt} = \frac{g}{l}\,y \tag{191}$$

$$(192)\qquad \frac{dy}{dt}=\frac{A}{F}(v_1-v)$$

Dividing (192) by (191):

$$\frac{dy}{dv}=\frac{Al}{Fg}\cdot\frac{v_1-v}{y}$$

Integrating:

$$(193)\qquad \frac{y^2}{2}=\frac{Al}{Fg}\left(v_1v-\frac{v^2}{2}\right)+C$$

To determine the constant of integration C; let $v=v_0$ when $y=o$, whence:

$$C=\frac{Al}{Fg}\left(\frac{v_0^2}{2}-v_1v_0\right)$$

Substituting this value in (193) gives:

$$(194)\qquad y^2=\frac{Al}{Fg}\left[(v_1-v_0)^2-(v_1-v)^2\right]$$

Substituting this value of y in (191) and solving for dt gives:

$$(195)\qquad dt=\sqrt{\frac{lF}{Ag}}\cdot\frac{dv}{\sqrt{(v_1-v_0)^2-(v_1-v)^2}}$$

The integral of (195) is:

$$(196)\qquad t=-\sqrt{\frac{lF}{Ag}}\sin^{-1}\frac{v_1-v}{v_1-v_0}+C$$

When $t=o$, $v=v_0$, hence

$$C=\frac{\pi}{2}\sqrt{\frac{lF}{Ag}}$$

after which (196) becomes:

$$t=\sqrt{\frac{lF}{Ag}}\left[-\sin^{-1}\frac{v_1-v}{v_1-v_0}+\frac{\pi}{2}\right]\text{ or}$$

$$t=\sqrt{\frac{lF}{Ag}}\cos^{-1}\frac{v_1-v}{v_1-v_0}$$

Solving this equation for v gives:

$$(197)\qquad v=v_1-(v_1-v_0)\cos\sqrt{\frac{Ag}{lF}}\,t$$

If this value of v be now substituted in equation (192) the equation for y in terms of t can be obtained as follows:

$$\frac{dy}{dt} = \frac{A}{F}(v_1 - v_0)\cos\sqrt{\frac{Ag}{lF}}\,t$$

$$y = \frac{A}{F}(v_1 - v_0)\sqrt{\frac{lF}{Ag}}\sin\sqrt{\frac{Ag}{lF}}\,t + C$$

When $y = o$, $t = o$, hence $C = o$ and

$$(198) \qquad y = \sqrt{\frac{Al}{Fg}}(v_1 - v_0)\sin\sqrt{\frac{Ag}{lF}}\,t$$

Since this equation is that of a true sine curve it will be readily seen that the maximum ordinate and hence the maximum departure of the head from normal is

$$(199) \qquad Y = \pm\sqrt{\frac{Al}{Fg}}(v_1 - v_0),$$

and return to normal head occurs when

$$\sqrt{\frac{Ag}{lF}}\,t = \sqrt{\frac{Ag}{lF}}\,T = \pi$$

Whence

$$(200) \qquad T = \pi\sqrt{\frac{lF}{Ag}}$$

Equations (197) and (198) may now be revised to read

$$(201) \qquad v = v_1 - (v_1 - v_0)\cos\frac{\pi}{T}t \text{ and}$$

$$(202) \qquad y = Y\sin\frac{\pi}{T}t$$

These equations (201) and (202), are shown for a particular problem, by the dotted lines B_v and B_h in Figure 285. The closeness of their agreement with the curves A_v and A_h which involve the effect of both friction and governor action shows that the values T and Y would commonly be as close to the truth as the estimate could be made of the probable load change $(v_1 - v_0)$, for which the stand-pipe should be designed.

More exact formulas can be derived, however, from the standpoint of energy as follows:

Let the time required to reach D' and hence to approximately reach the valve v_1, under exact conditions, be $\frac{T'}{2}$.

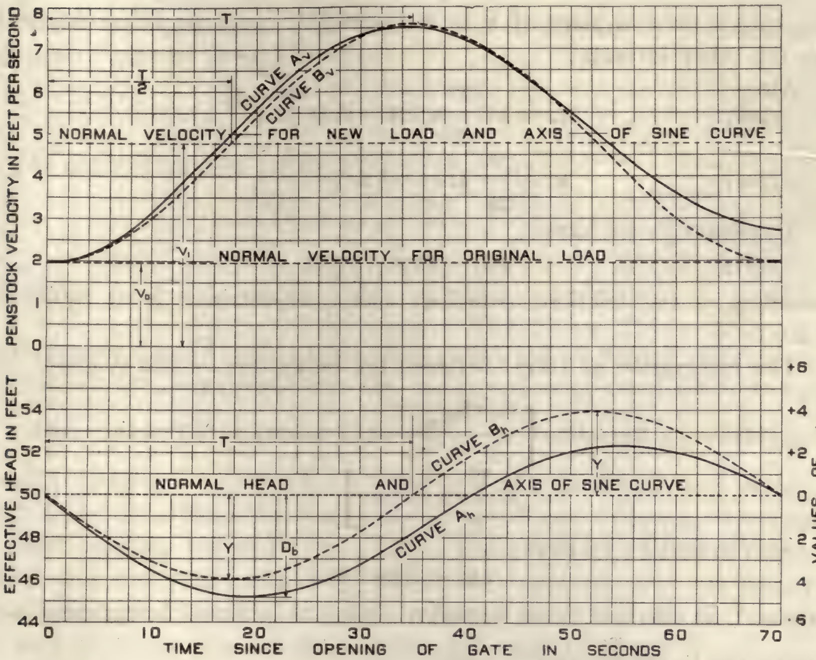

Fig. 285.—Curves Showing Fluctuations of Head and Penstock-Velocity in a Plant With Stand-pipe (see page 450).

The time $\frac{T'}{2}$ will be slightly greater than $\frac{T}{2}$, when friction and governor action are involved, and the method of determining it will be given later (equation 213).

It is evident that the number of foot pounds of energy which must be supplied by the stand-pipe in this time $\frac{T}{2}$ is equal to the energy required by the wheel plus that required to accelerate the water in the penstock plus that necessary to overcome the friction of the penstock minus that supplied through the penstock,

(203) Or $$E_s = E_w + E_a + E_f - E_p$$

Now,

(204) $$E_s = wFD' \left(H - cv_0^2 - \frac{D'}{2} \right)$$

where D' is the maximum surge below the initial friction gradient for

V_0, and is used in place of Y to distinguish it from the value obtained by the other formula:

Also,

$$(205) \qquad E_w = Av_1 \frac{T'}{2} w\,(H - cv_1^2) \text{ and}$$

$$(206) \qquad E_a = \frac{w}{2g} Al\,(v_1^2 - v_0^2)$$

To obtain E_f we have

$$(207) \qquad dE_f = Avw \times cv^2\,dt$$

where c is the friction coefficient and v is obtained from equation (201).

The integration of (207) between the limits $t = \frac{T'}{2}$ and 0, gives,

$$(208) \qquad E_f = Awc \left[\frac{v_1^3 T'}{2} - \frac{3T'}{\pi} v_1^2 (v_1 - v_0) + \tfrac{3}{4} T' v_1 (v_1 - v_0)^2 - \frac{T'}{6} (v_1 - v_0)^3 \right]$$

Also to find E_p we have

$$dE_p = HAwv\,dt,$$

where v is obtained from equation (201) as before. Integrating between the limits $\frac{T'}{2}$ and 0, gives

$$(209) \qquad E_p = HAwT' \left(\frac{v_1}{2} - \frac{v_1 - v_0}{\pi} \right)$$

Combining and simplifying:

$$(210) \qquad D'^2 - 2\,(H - cv_0^2)\,D' = -\frac{2A}{F} \left\{ \frac{1}{2g} (v_1^2 - v_0^2) + c \left[-\frac{3T'}{\pi} v_1^2 (v_1 - v_0) + \tfrac{3}{4} T' v_1 (v_1 - v_0)^2 - \frac{T'}{6} (v_1 - v_0)^3 \right] + \frac{HT'}{\pi} (v_1 - v_0) \right\}$$

$$(211) \qquad D_b = D' + cv_0^2$$

The upward surge can be found by the same equation by a proper change of signs, but is unimportant since it is always less than the downward surge D_b for the same change of velocities.

If friction be omitted and T' be changed to T for reasons mentioned later, equation (210) reduces to

$$(212) \qquad D^2 - 2HD = -\frac{2A}{F} \left\{ \frac{1}{2g} (v_1^2 - v_0^2) + \frac{HT}{\pi} (v_1 - v_0) \right\}$$

To derive an equation for the maximum upward surge D_a, when full load is rejected, we may equate the original kinetic energy in the penstock to that expended in friction plus that used in raising

water in the stand-pipe. The energy lost in friction is found from equation (208) by putting $v_1 = o$

$$\text{or } E_f = \frac{AwcTv_0^3}{6}$$

The other quantities are evident. This gives:

$$\frac{WAL}{2g} v_0^2 = \frac{AwcTv_0^3}{6} + \frac{wFD_a^2}{2}$$

or—

(213) $$D_a^2 = \frac{A}{F} v_0^2 \quad \frac{1}{g} - \frac{cTv_0}{2}$$

Equations (205), (208), (209) and (210) are all theoretically exact except for the assumption that the velocity change takes place along a simple harmonic in time $\frac{T'}{2}$. The true curve for a half cycle, as used, is scarcely distinguishable from a simple harmonic but its period T' or time for return of water in stand-pipe to normal level is greater than the value T, given by equation (164). In three cases which the writer has solved by successively applying the differential equations to short positions of the arc he has found that the true value T' may be closely approximated by the following formula:

(214) $$T' = \frac{D}{Y} T$$

where T is found from equation (200),

Y from equation (199), and

D from equation (212).

The quantity T', is useful in itself as the true time for return to normal head, but its use in formula (210) for determining D' is not advisable, as the writer has found by solving a number of problems that the value of D', thus found, agrees almost exactly with the value of D found from equation (212), in which equation the value of T from equation (200) is used. Equation (212) is therefore offered as a much simpler substitute for equation (200) and equation (211) becomes:

(215) $$D_b = D + cv_0^2 \text{ *}$$

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* Mr. Raymond D. Johnson in Am. Soc. M. E. 1908 has derived an equation for D as follows:

$$D'^2 = \frac{Al}{Fg} (v_1 - v_0)^2 + c^2 (v_1^2 - v_0^2)^2$$

The results obtained by this equation agree quite closely with those obtained by the writer's method and the two entirely independent analyses of the problem are mutually corroborative.

Like all wave motions, these surge waves are liable to pile up, one upon another, in case several gate movements occur at proper intervals and, in fact, no limit can be placed upon the possible amplitude of the surge which can occur in this way. In a plant where large frequent load changes are anticipated the danger from this source should receive careful attention. Some means should be adopted for causing the wave, due to a given gate movement, to rapidly subside in order to lessen the probability of its combination with another wave. One method of accomplishing this result is by arranging the stand-pipe to overflow at a definite elevation above the forebay. This limits the upward surge and thereby the maximum possible downward surge which could occur under any assumption of gate movements. This method necessitates a waste of water.

Another method * is that of imposing a resistance between penstock and stand-pipe. This not only causes the waves to subside more rapidly but also, if properly designed, reduces the amplitude of a single wave. This is of greatest advantage near full load where the downward surge is apt to lower the head sufficiently to make it impossible for the unit to deliver the required power. Another effect of the resistance, however, is to change the form of the curve of effective head so that, instead of a slow sinuous pressure drop after an increase of load, a sudden drop is obtained. This is evidently opposed to good speed regulation as it adds to the effective sudden load for which the governor must compensate by requiring a greater q to make up, not only for the increased load, but also for the suddenly decreased head.†

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* See paper on "Surge Tanks for Water Power Plants" by R. D. Johnson with discussions by the writer and others in the Trans. Am. Soc. of M. E. 1908.

† For further discussion of this subject and a mathematical analysis of the problem see Mr. R. D. Johnson's paper with discussions as previously referred to.

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CHAPTER XV

THE WATER WHEEL GOVERNOR

215. Types of Water Wheel Governors.—In all reaction turbines and in all impulse turbines, with the exception of tangential wheels, the governor affects regulation, i. e., controls the output, and hence the speed of the wheel, by opening or closing the regulating gates, thus varying the amount of water supplied to the wheel. In tangential wheels, under high head, this method of control, for obvious reasons (see Section 196), becomes difficult and in extreme cases impossible and in such cases the governor must be arranged to affect regulation by the deflection of the jet from the bucket (see Fig. 286).

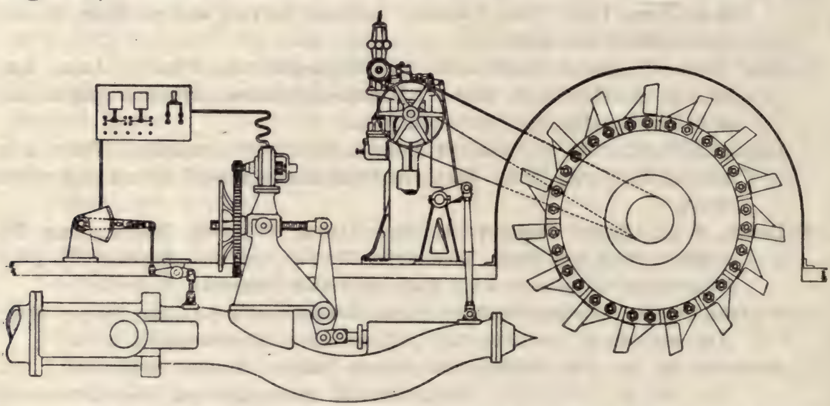

Fig. 286.—Governing Impulse Wheel with Automatic Needle and Deflecting Nozzle (after Warren).

The force required to move the turbine gates is large (sometimes 50,000 lbs. or more) and it is therefore evident that they cannot be moved by the direct action of the centrifugal ball governors, as with steam engines, but must be moved by a "relay."

The relay, as its name implies, is a device for transmitting energy from a source of energy independent,—as to quantity—of the centrifugal governor balls but controlled by them in its application. If the relay is of *"mechanical type"* the power required to operate it and the gates is transmitted, when needed, from the wheel by

means of shafts, gears, friction-clutches, belts and pulleys or other mechanical devices. In mechanical governors the flyballs may actuate pawls, friction gears or other mechanical devices which will bring the relay into action.

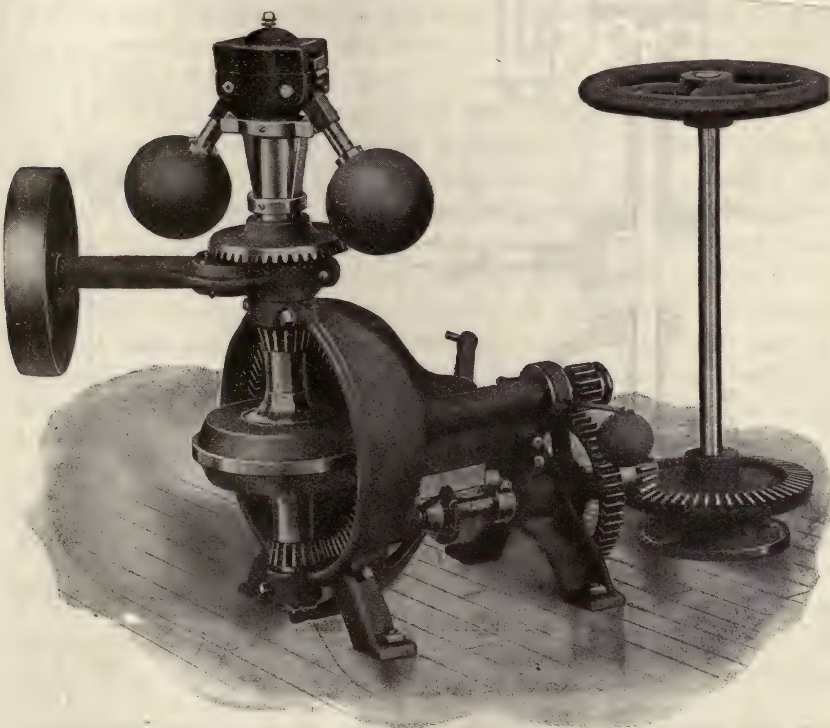

Fig. 287.—Woodward Standard Governor (see page 462).

If the relay is of the *hydraulic type,* it usually consists of a piston connected by some mechanical device to the gate rigging and moved by means of the hydraulic pressure of water taken from the penstock, or other source, or by oil supplied under high pressure from a reservoir. The pressure of the oil in the reservoir is maintained by compressed air supplied by power taken from the wheel itself. The oil thus used in moving the piston is exhausted into a receiver from which it is pumped back into the supply reservoir. The hydraulic relay is commonly controlled by the ball governor through the medium of a small valve which by its motion either admits the actuating water (or oil) directly to the cylinder or to a secondary piston controlling a larger admission valve.

Electrical methods of actuating the relays controlled by means of governor balls have been used to some extent but are not nearly so common as mechanical or hydraulic devices.

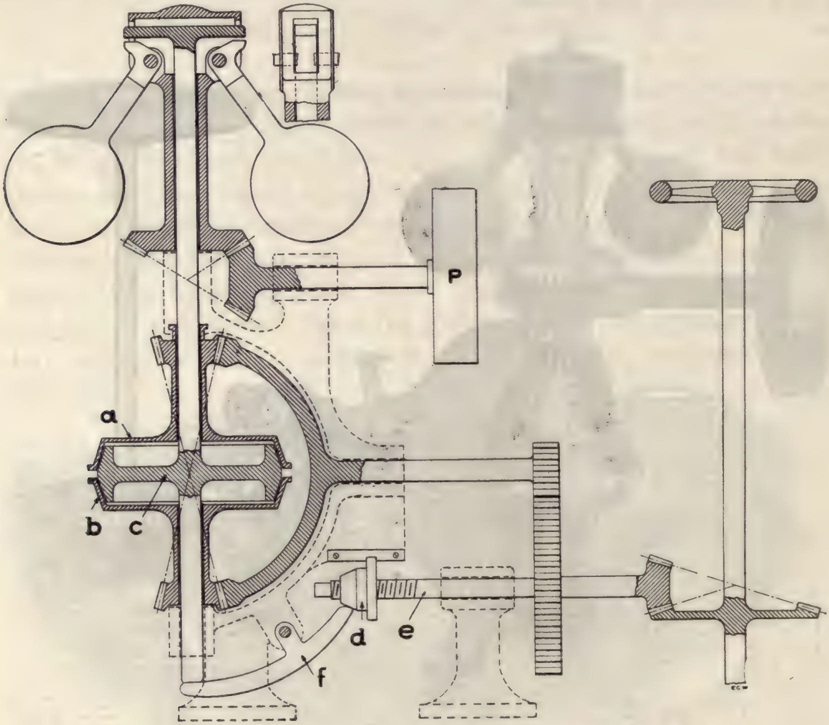

Fig. 288.—Diagramatic Section of Woodward Simple Mechanical Governor.

216. Simple Mechanical Governors.—Fig. 287, page 461, is a view and Fig. 284 a diagramatic section of a simple mechanical governor of the Woodward * Standard type. On the upright shaft are two friction pans *a* and *b* (see also Fig. 291, page 466). These pans are loose on the shaft, the upper one being supported in position by a groove in the hub and the lower one by an adjustable step-bearing. Between these pans, and beveled to fit them, is a double-faced, friction wheel *c*, which is keyed to the shaft. This shaft and friction wheel run continuously and have a slight endwise movement. They are supported by lugs on the ball arm and therefore rise and fall as the position of the balls varies with the speed.

* Woodward Governor Co., Rockford, Ill.

When the speed is normal, the inner or friction wheel revolves freely between the two outer wheels or pans which remain stationary. When a change of speed occurs, the friction wheel is brought against the upper or lower pan as the speed is either slow or fast. This causes the latter to revolve and, by means of the bevel gearings, turn the gates in the proper direction until the speed is again normal. As the gate opens the nut *d,* travels along the screw *e,* which is driven through gearing by the main governor shaft and as the gate reacts, the nut *d,* coming in contact with the lever *f,* throws the vertical shaft upward and the governor out of commission.

This type of governor may be used to advantage where the water wheels operate a number of machines, connected to a main shaft and where, in consequence, the friction or constant load is a considerable percentage of the total load. In such cases the changes in load may not be a large percentage of the total load and the temporary variations in speed, which occur at times of changes of load, may not be of sufficient importance to necessitate the installation of a quick acting governor.

When the water wheel is direct connected to a single machine, and the friction load is comparatively small, the relative change in load, and the consequent possible changes in speed, is much larger.

In such cases the type of governor above shown will result in a serious hunting or racing (see Section 193) of the wheel during considerable changes of load, and in unsatisfactory regulation. In such cases governors with compensating or anti-racing devices must be used for satisfactory regulation.

217. **Anti-Racing Mechanical Governors.**—The Woodward compensating governor—Fig. 289, page 464, is a view and Fig. 290, page 465, is a diagramatic section of a Woodward vertical mechanical governor of the compensating type.

In the simple Woodward governor (see Figs. 278, page 461, and 288, page 462) the power necessary to actuate both the centrifugal governor balls and the relay is transmitted through a belt to a single pulley *P.* In the Woodward compensating type of governor the relay is operated in a similar manner by a single pulley *P,* while the centrifugal governor balls are actuated by an independent pulley *q,* having an independent belt connected to the wheel shaft or to some other revolving part connected therewith. From the driving pulley *q,* power is transmitted to the governor ball through a shaft and gearing. The shaft supporting the centrifugal governor balls is hollow, and on

the ball-arms are two lugs which connect with a spindle *f*, which therefore rises and falls as the positions of the governor balls vary with the speed.

The movement of the centrifugal governor balls causing the spindle *f*, to rise and fall changes the position of the tappet arm *g*. to

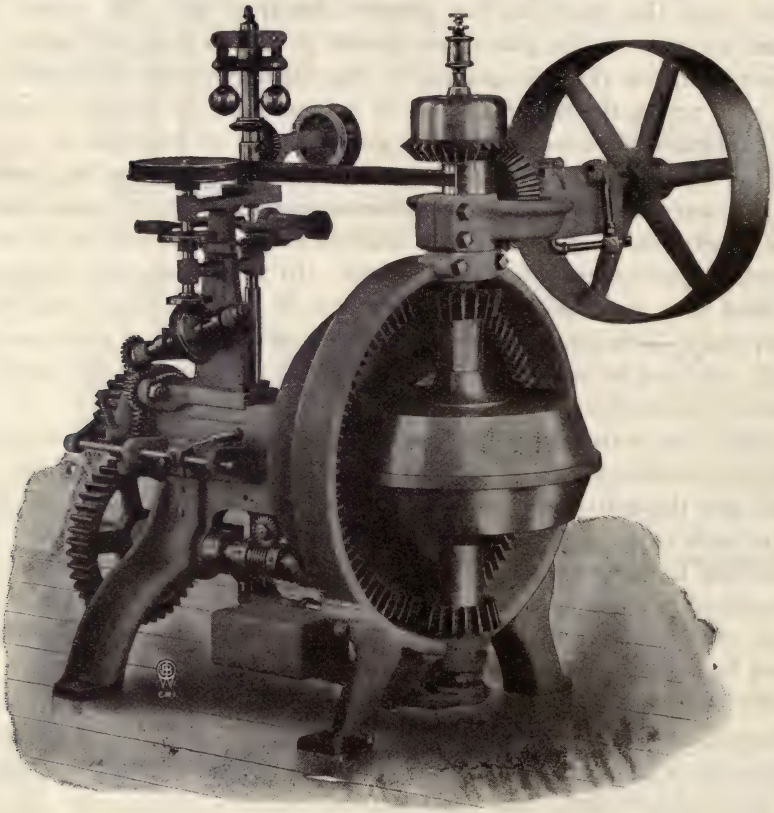

Fig. 289.—Woodward Compensating Governor (see page 463).

which it is connected, and causes one or the other of the two tappets *tt'*, to engage a double-faced cam *h*. This cam is continuously rotated by means of the pulley above it, driven by a belt connected with the main vertical shaft of the relay. The tappets are connected to a common suspension arm to which the vertical spindle *f*, is attached. The suspension arm is hinged to the lever arm, *j*. The lever arm is connected to the shaft *K*, which can be rotated on its bearings and which is connected with a tension rod *l*, by an eccentric at the bottom. The

tension rod *l*, is in turn connected by a lever *m*, with the vertical bearing *e*, on which the main shaft of the friction cone rests. This bearing is movable around the fulcrum *n*, and is counterbalanced by an arm and weight *u*.

When either of the tappets engages the rotating cam, the resulting movement turns the rocker shaft *K*, and, through its connection, raises or lowers the vertical bearing *e*, which causes the friction wheel *c*, to

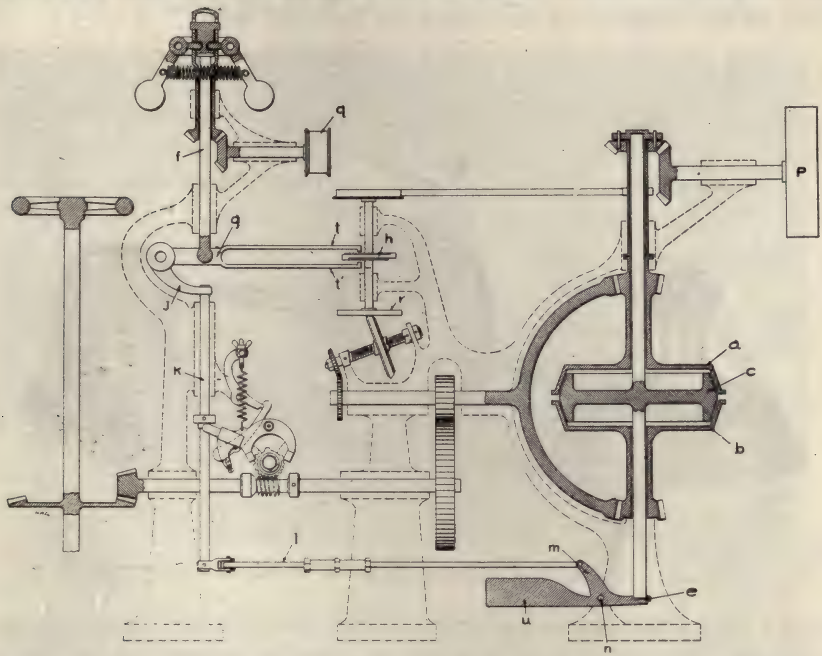

Fig. 290.—Diagramatic Section of Woodward Vertical Compensating Mechanical Governor (see page 463).

engage either the upper or the lower of the friction pans *a* and *b*, as in the case of the simple governor.

The compensating or anti-racing mechanism is just below the rotating cam. It is essentially alike in all of the Woodward compensating types of governors and is described in the governor catalogue as follows:

"On the lower end of the cam shaft is a friction disc *r* (Fig. 290) which rests on a rawhide friction wheel on a diagonal shaft. The hub of the friction wheel is threaded and fits loosely on the diagonal shaft which is normally at rest. The effect of the continually rotating friction disc upon the rawhide wheel is evidently to cause it to travel along

the threaded diagonal shaft to the center of the disc. When the governor moves to open or close the gate, the diagonal shaft, which is geared to it, is turned and the friction wheel is caused to travel along the shaft away from the center of the disc and thus raise or lower the cam shaft so as to separate the cam from the tappet which is in action, before the gate has moved too far, thus preventing racing. As soon as the gate movement ceases the disc causes the friction wheel to return to the center of the disc along the threaded shaft."

To prevent the governor from straining when the gate is fully open or closed, suitable cams are mounted on the stop shaft. "When the gates are completely opened, the cam engages the speed lever and holds it down so that it cannot raise the lower tappet sufficiently to engage the revolving cam; this does not, however, interfere with the

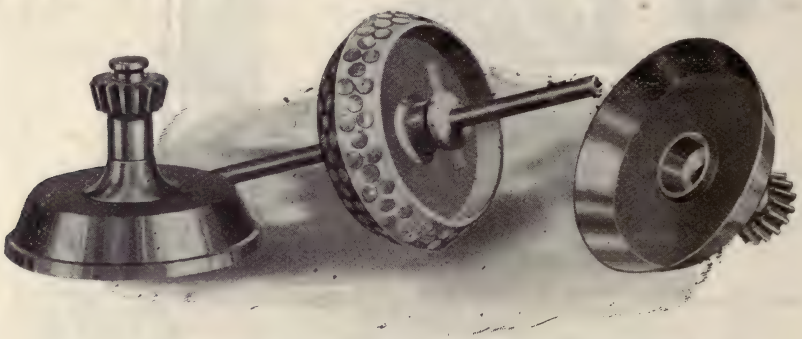

Fig. 291.—Friction Cone and Pans of Woodward Governor.

upper tappet, to prevent the closing of the gates, should the conditions demand. The closed gate stop acts in a similar manner on the upper tappet but does not interfere with the lower tappet being engaged, should the conditions demand that the gate be opened. In addition to these stops, the governor is provided with a safety stop whose function is to immediately close the gates should the speed governor stop through breakage of the belt or any other cause."

218. Details and Applications of Woodward Governors.—Figure 291 shows the construction of the friction gearing of the Woodward mechanical governor. In the inner friction driving cone, corks are inserted in holes drilled in the rim and these are ground off true so that they project about one-sixteenth inch. This seems to give a very reliable friction surface not readily affected by either water or oil, and it is claimed to be superior to either leather or paper for this

purpose. In order to cause the friction wheel to engage smoothly and noiselessly, a plunger attached to the shaft, just below the inner friction wheel fits rather closely into a dash-pot formed in the lower pan.

Figure 292 shows an horizontal compensating type of Woodward governor as installed to control the gates of the turbines in the Hydraulic Power Plant of the United States Arsenal at Rock Island, Illinois. The cables shown at the back of the cut operate the gates of the tur-

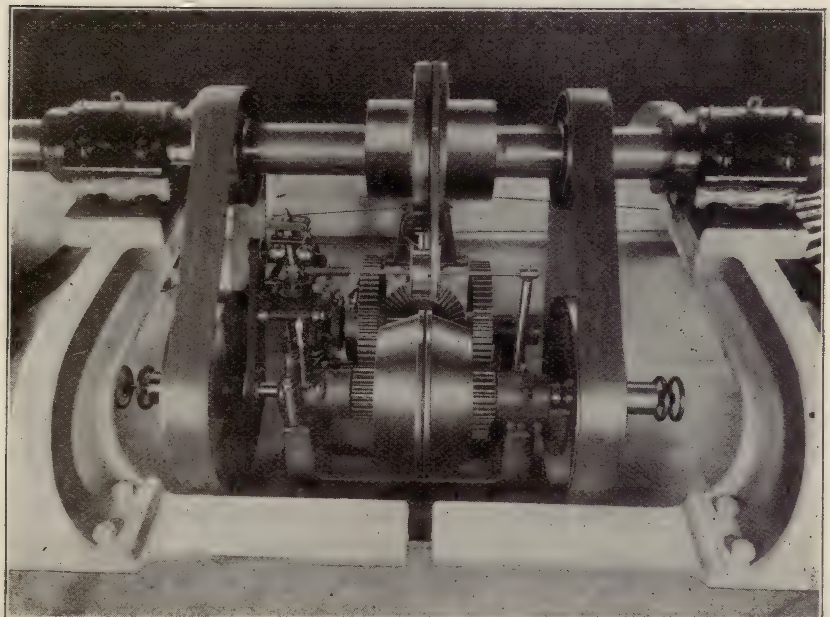

Fig. 292.—Woodward Horizontal Compensating Mechanical Governor at Hydro-Electric Plant of U. S. Arsenal, Rock Island, Ill.

bine. On the gate shafts of the latter are sheave wheels to which the cables are attached. These sheave wheels are fitted with clutches so that any gate may be disconnected from the governor. Each gate is provided with an indicator showing its position. This provides means of coupling properly, after being disconnected, without closing the gates of the other wheels. Each governor is arranged to control six turbines, belonging to two different units. Two belts are provided so as to drive from either unit. The governor can thus be used to control three wheels on either side or all six when the two units are running in multiple.

219. Essential Features of an Hydraulic Governor.—The essential features of an hydraulic water wheel governor are:

1. A tank for storing oil under air pressure.
2. A receiver tank for the collection of oil used by the governor.
3. A power pump driven from the water wheel shaft.
4. An hydraulic power cylinder for operating the gates.
5. A sensitive centrifugal ball system for controlling a valve which either admits oil directly to the power cylinder or to an intermediate relay cylinder the piston of which operates the admission valve to the power cylinder.
6. An anti-racing or compensating mechanism.

The power pump is continually using power from the wheel to pump the oil from the receiver back to the pressure tank, thus gradually storing the energy which is used intermittently to operate the gates.

Fig. 293, page 469, illustrates the Lombard Type "N" governor and shows clearly the relations of the various parts of an hydraulic governor.

The centrifugal governor balls are connected by belt to the wheel shaft. These balls control a small primary or pilot valve of the cylinder type which admits oil from the large pressure tank under about 200 pounds pressure into one side of a cylinder where its pressure is exerted against one of two plungers. These plungers control a large valve, also of the cylinder type, which admits oil from the pressure tank to one or the other side of the power piston. The rectilinear motion of the piston is converted by rack and pinion, into rotary motion for transmission to the wheel gates. The oil used for operating the power pistons and the plungers of the relay is exhausted into the vacuum tank from which it is pumped back into the pressure tank by means of the power pump shown at the left which is driven by belt from the wheel shaft. The speed variation necessary to actuate the governor depends upon the lap of the pilot valve and is adjustable.

220. Details of Lombard Hydraulic Governor.—The details of the Lombard Type "N" governor are best shown by the enlarged view of the upper portion of the governor (Fig. 294, page 470) and by the section of the relay valve (Fig. 295, page 471). The following description of the operation of this governor is taken from the Directions for Erecting and Adjusting Governors.*

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* Published by The Lombard Governor Co., Ashland, Mass.

"The oil from the pressure-tank is supplied to the working cylinder 62 through the large relay-valve 106, arranged to discharge or exhaust oil directly and rapidly into or from either end of the cylinder. The relay-valve 106, through the hydraulic system connected therewith, is

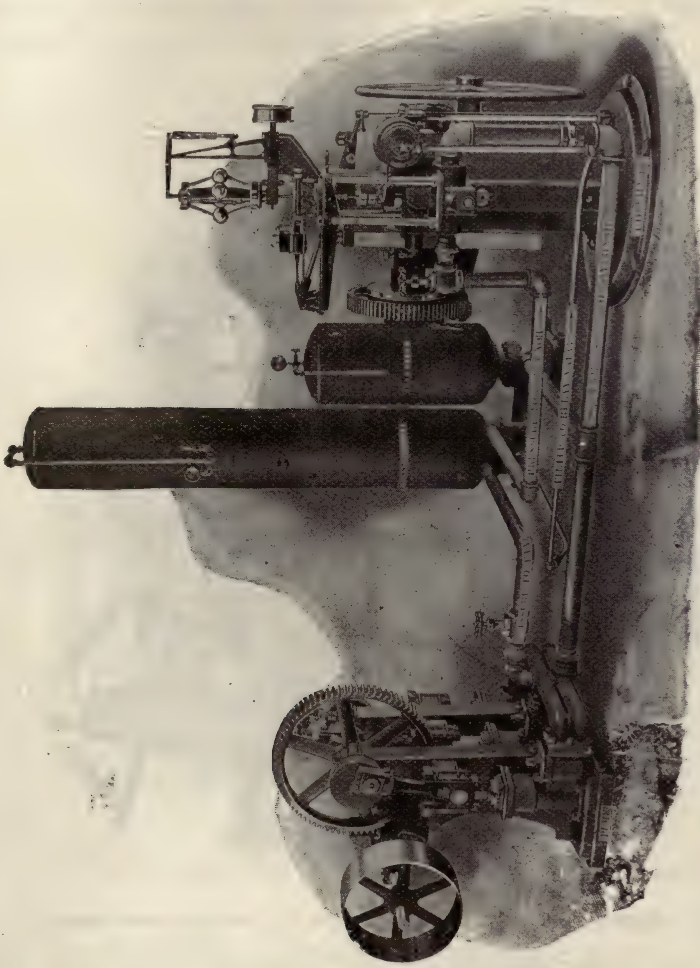

Fig. 293.—Lombard Type N Hydraulic Governor (see page 468).

under the simultaneous control of the regulating-valve 14 and the displacement-cylinder 107. This is brought about in the following manner. The relay-valve A (see Fig. 295, page 471), is moved hydraulically by plungers B and C, contained within cylinders D and E, forming parts of the relay-valve heads F and G. Plunger B, has about one-half the area of plunger C, consequently plunger C, can overpower

plunger *B,* if the pressure in cylinders *E* and *D* is nearly equal. The cylinder *D,* is permanently in communication with the main pressure supply through the pipe *H,* which also furnishes liquid to the regulating-valve 14. Therefore the tendency of plunger *B,* is always to move

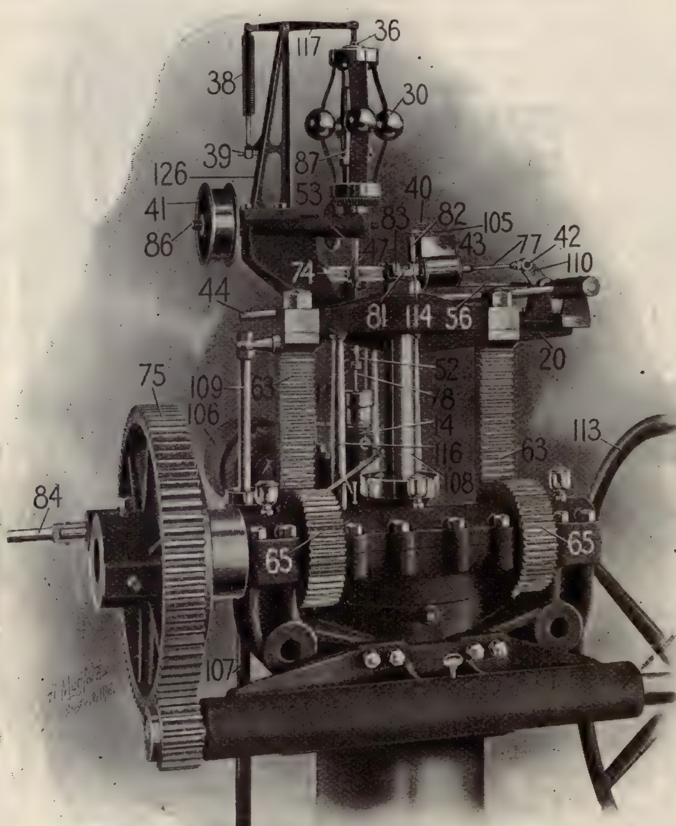

Fig. 294.—Upper Portion of Lombard Type N Governor (see page 486).

valve *A,* towards the relay-valve head *G.* Cylinder *E,* is in communication through pipes *I* and *J,* with the adjusting-valve 14, and also through the pipes *J* and *K,* with the displacement-cylinder 107. The regulating-valve 14 is capable, when moved in one direction, of admitting liquid under full pressure into the pipe, *I,* and, when moved in the other direction, of exhausting liquid through the pipe *I.* In the former

case the action is to increase the pressure back of the piston *C*, until it overpowers the piston *B*, thereby moving valve *A*, towards the relay-valve head *F*, simultaneously opening the upper cylinder-port to the main exhaust, and the lower cylinder-port to the main pressure supply. Instantly the main piston of the governor and with it the displacement-plunger 109 are set in motion.

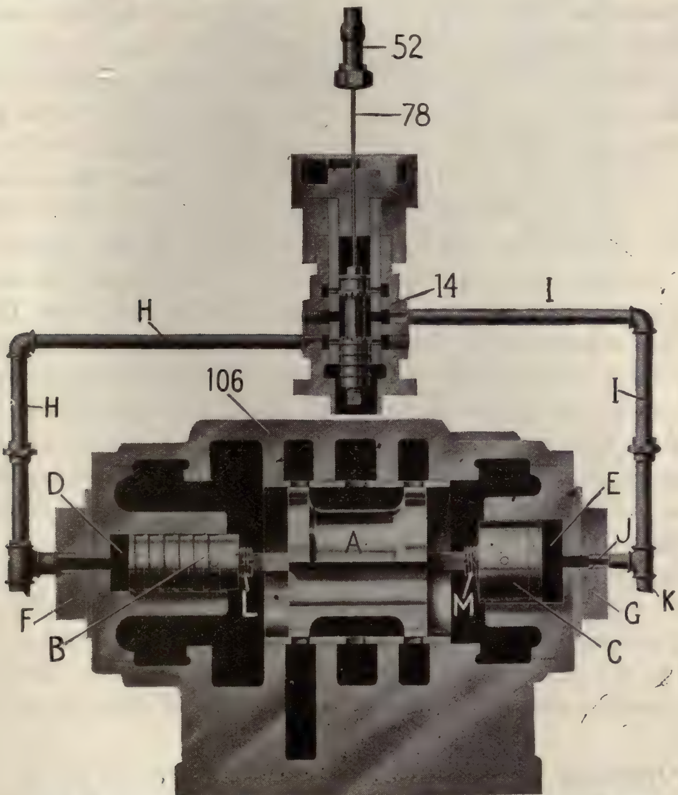

Fig. 295.—Section Lombard Delay Valve (see page 486).

"As the displacement-plunger begins to move, a space is created back of it, into which a portion of the liquid flowing through the pipe *I*, is diverted. As the motion of the displacement-plunger becomes more rapid, a condition is reached when all the liquid flowing through *I* continues on through *K* into the displacement-chamber. The relay-valve *A*, then ceases to move any further. The motion of the main governor-piston, however, continues as long as the regulating-valve 14 is open. When this valve 14 closes, the relay-valve *A*, is immediately

thereafter closed, because the liquid in the cylinder *E*, instantly escapes through the pipes *J* and *K*, into the space beneath the moving displacement-plunger; thus the whole governor is brought to rest.

"When the regulating-valve 14, is moved in the opposite direction by the centrifugal balls so as to allow liquid to escape through the pipe *I*, there results an immediate loss of liquid in the cylinder *E*, back of the plunger *C*; this allows the plunger *B*, to force the relay-valve *A*, towards the relay-valve head *G*, thus opening the lower cylinder port to the exhaust, and the upper cylinder-port to the pressure supply. The main governor-piston instantly begins to move down, carrying with it the displacement-plunger, thus forcing liquid through the pipes *K* and *I*, reducing the flow outward through *J*, until finally the downward velocity of the displacement-plunger becomes rapid enough to entirely check the outward flow through *I*. Relay-valve *A*, then remains stationary until the valve 14, has moved to a new position. As soon as regulating-valve 14, is closed, the liquid which has been flowing out through *I* immediately flows into *J* and, acting upon the plunger *C*, restores valve *A*, to its closed position, stopping further movement of the governor. It will be seen that the governor when moving has a constant tendency to close the relay-valve which keeps it in motion, and this relay-valve can be maintained open only so long as the regulating-valve 14, is adding or subtracting oil to or from the system consisting of the pipes *I*, *J*, *K*, and parts connected therewith."

Figure 296, page 473, shows the Lombard Governor Type "R", the smallest of the various governors made by that company. This is a vertical, self-contained oil pressure machine. The oil is stored in a tank formed by the main frame. The governor is designed to exert 2,500 pounds pressure and will make an extreme stroke of eight inches in one second.

221. Operating Results with Lombard Governor.—Fig. 297, page 474, is a cut from a speed recorder strip taken from the Hudson River Power Transmission Company's plant and shows the regulation of the Lombard Type "B" governor regulating S. Morgan Smith turbines on an electric railroad load. The cars are large and the change in load rapid and large.

Figure 298, page 475, shows the comparative regulation of two generators in the same plant (see Bulletin No. 107 Lombard Governor Company). The load was quite variable on account of beaters which had to be driven from the same shaft as the paper making machinery. The

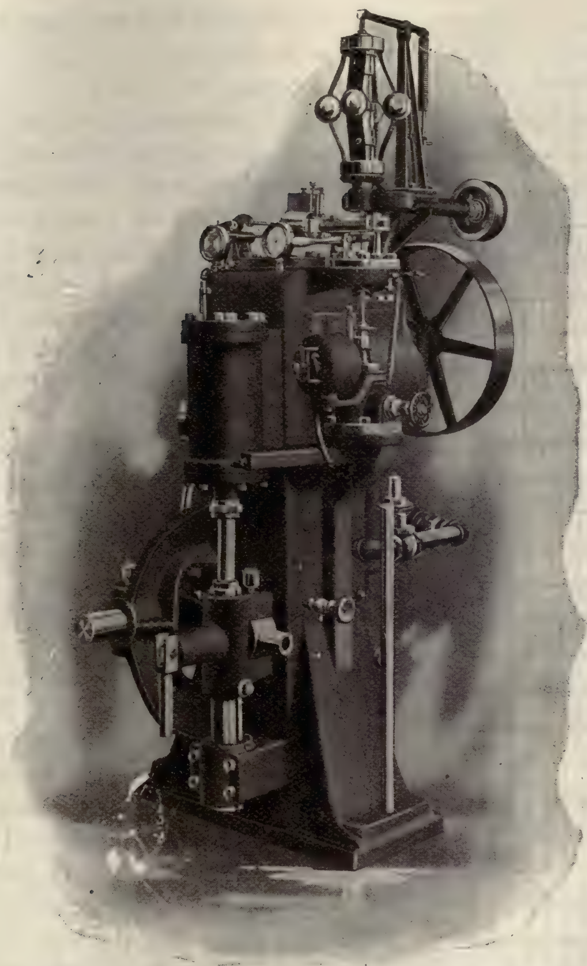

Fig. 296.—The Lombard Type R Governor (see page 472).

original governor used, the work of which is shown up the upper cut, was replaced by a Lombard Type "D" governor. The work of the latter is shown in the lower tachometer chart, and the improvement in the uniformity of operation is readily seen by a comparison of the two charts.

222. The Sturgess Hydraulic Governor.*—The Sturgess Type "M" hydraulic governor, with the omission of the pump and storage

* Sturgess Engineering Dept. of The Ludlow Valve Mfg. Co., Troy, N. Y.

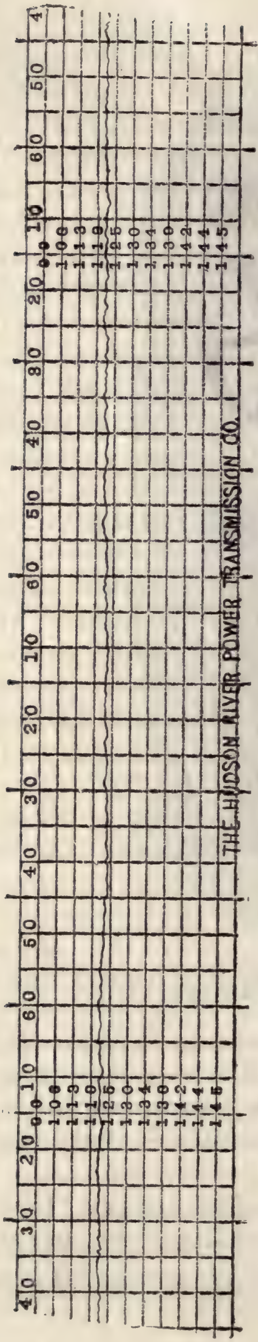

Fig. 297.—Speed Record from Plant of Hudson River Power Transmission Co. (see page 472).

tank, is shown in Fig. 299, page 476, and in section in Fig. 300, page 477. This governor consists of a shaft-type centrifugal governor *G*, attached to the top of the machine and operated by a belt and pulley *P*, from the turbine shaft. The governor balls *BB*, in this machine control directly by means of a long vertical lever *D*, a small primary or pilot valve *S*, of cylinder type which admits oil to a cylinder controlling the main admission valve *S*. The main valves, attached to the side of the cylinder, admit pressure directly into the cylinder *S*, and on either side of the piston *S*, which, by its motion, rotates the gate shaft by means of the concealed steel rack *R* and pinion *N*, shown in the sectional view, Fig. 300.

The valves for the admission of oil or water, as the case may be, in the cylinder are of the poppet type which avoid "lap" and therefore increase the sensitiveness of the governor. The anti-racing mechanism consists of a rod *A*. which is attached to the cross head of the governor. At the top of this rod is a projection to which is attached an adjustable piston rod reaching down into the open top dash pot *F*. The piston rod has a piston attached at its lower end fitting freely into the bore of the dash pot the top of which is formed into a cup which receives the excess oil. The bottom of the dash pot is closed and is attached to a tail piece connected to the counter weighted locker lever *C*.

The piston rod and piston are hollow and near the bottom of the piston is a small bypass which can be regulated by an adjusting screw which controls the rate of flow of the oil in the dash pot. The lever *C*, is fixed on the rocker shaft, the opposite end of which carries the short arm from which

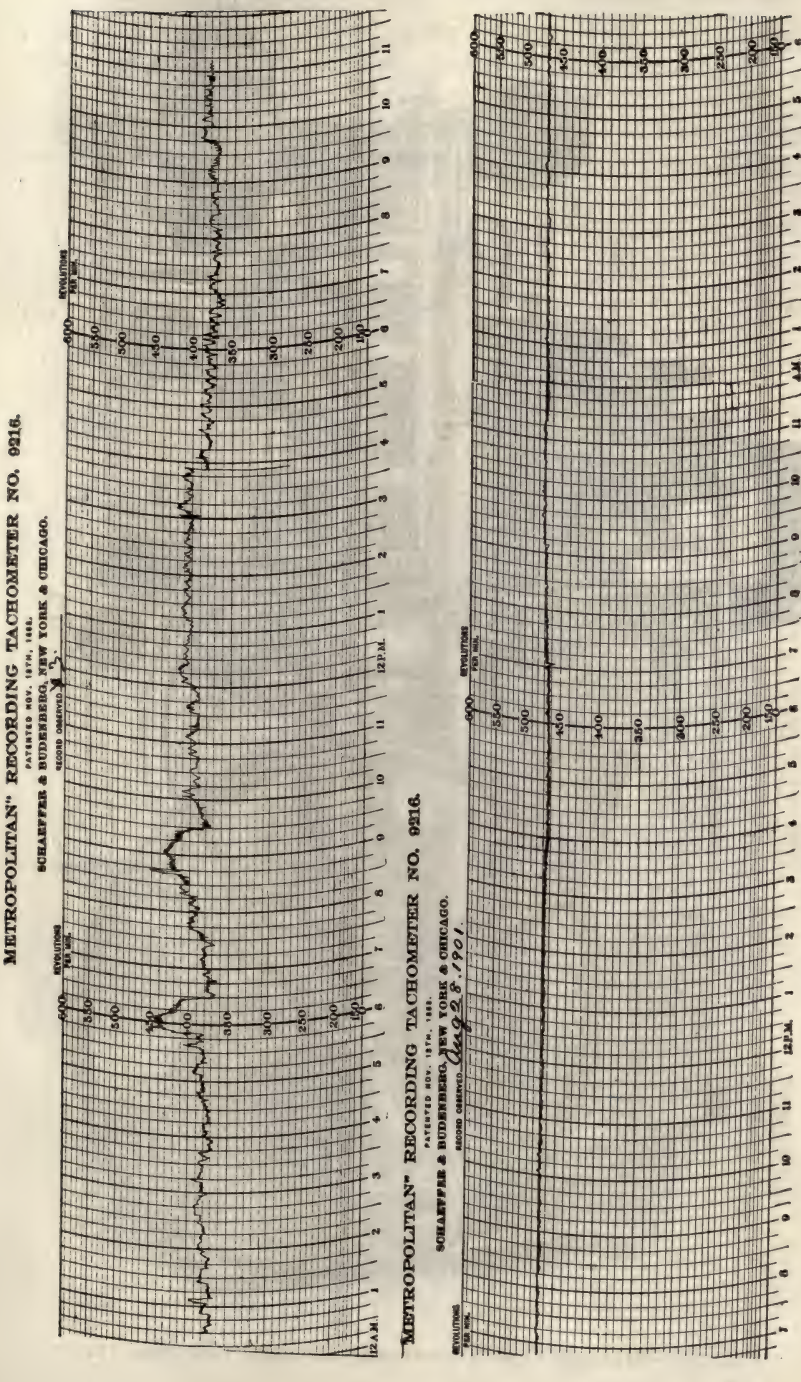

Fig. 298.—Comparative Turbine Regulation (see page 472).

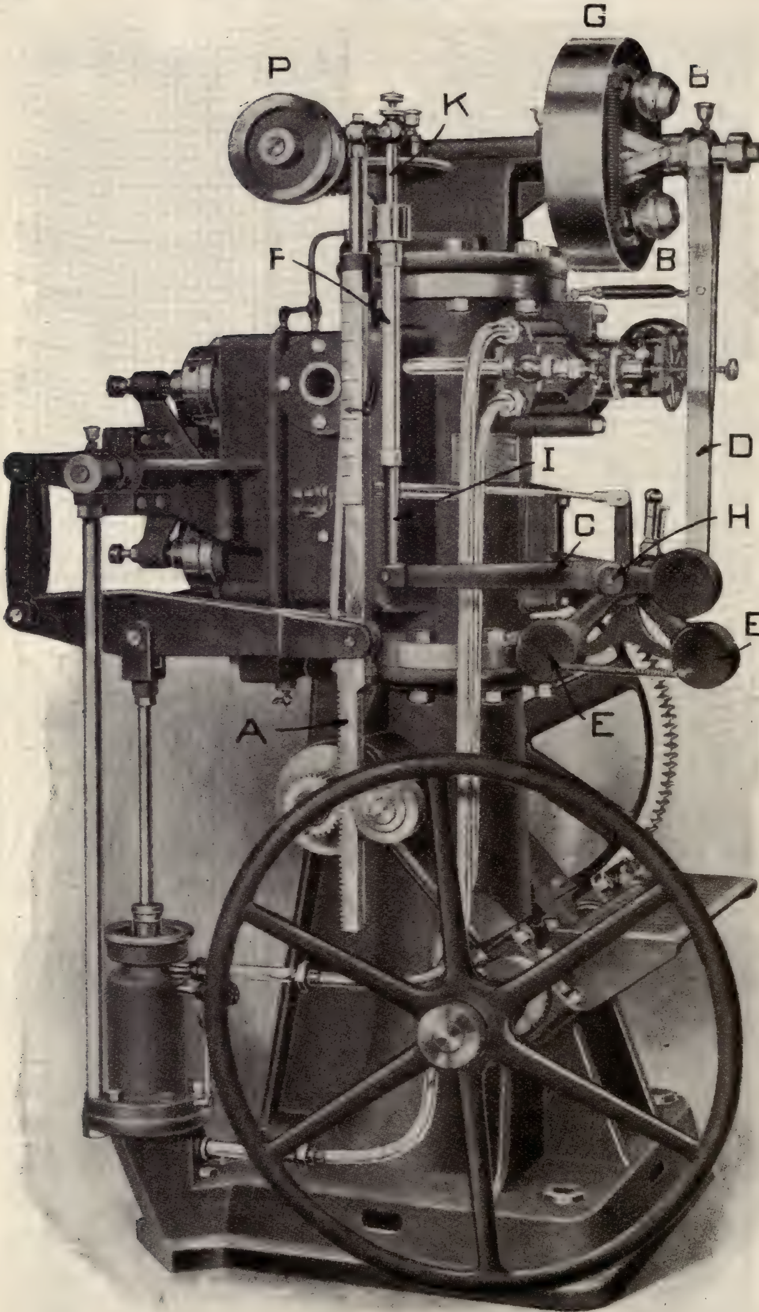

Fig. 299.—Sturgess Type M Hydraulic Governor (see page 474).

a link is carried to the bottom of the valve lever *D*, which is free to move. Two weights *EE*, are hung loosely on the rocker shaft but a pin on the shaft engages with either one or the other of the weights and raises them whenever the rocker shaft moves. The function of the weights therefore is to keep the rocker shaft, and consequently the

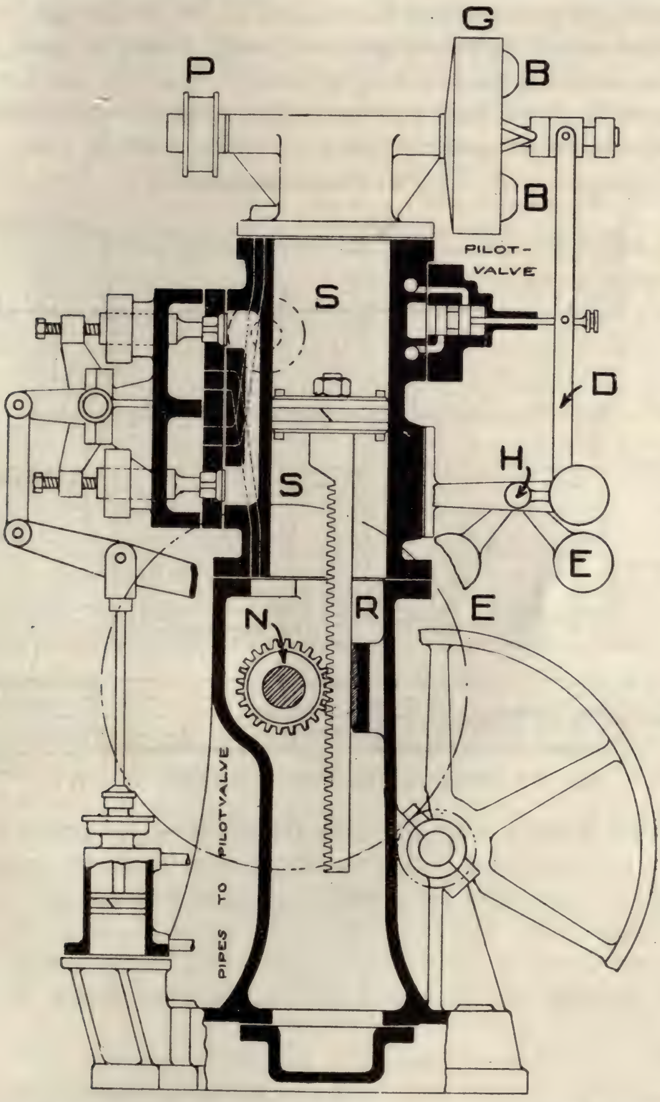

Fig. 300.—Section Sturgess Type M Governor (see page 474).

bottom of the valve lever, in normal position. When the main piston moves it is obvious that it will tend to raise or lower the dash pot *F*, through its connection to the rod *I*, and this movement will swing the lever *C*, and rocker shaft *H*, thus deflecting the bottom of the valve level *D*, so as to compensate in the correct manner. The same movement raises one of the weights *E*, but as the dash pot permits a slow movement the weights will finally restore all parts to the middle or normal position. In the smaller sizes the pilot valve is omitted and the centrifugal governor balls actuate directly through the lever the main valves of the system.

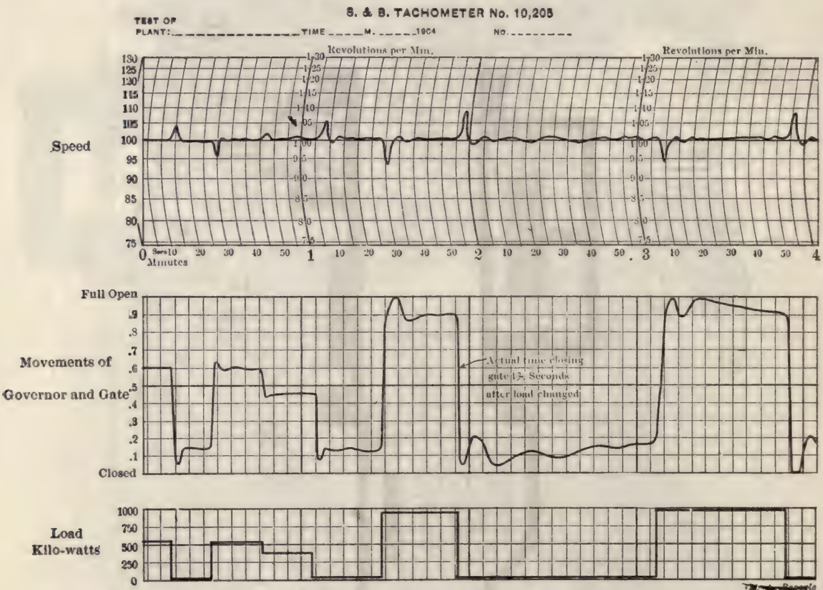

Fig. 301.—Test Results with Sturgess Governor.

223. **Test Results with Sturgess Governor.**—The action of any governor in maintaining a uniform speed may be shown graphically by attaching a recording tachometer to the turbine shaft. In order to fully understand and appreciate the action of the governor, the tachometer chart should be considered together with the load curve and a diagram showing the movement of the governor during the same period.

Figure 301 shows a governor test made by Mr. John Sturgess on an 1100 K. W. unit. "The curves were traced by a special Schafer & Budenberg tachometer, the readings being sufficiently mag-

nified to bring out the characteristics of the governor. * * * The load changes and governor movements are platted below. Note that when the whole load was thrown off (at 1:55), the speed accelerated about eight per cent. in an incredibly short time (under one second), and the governor had the gate shut in one and four-tenths seconds after the load went off. * * * It is to be noted that after the first quick result at 2:00 minutes the governor slowly oscillated for about another minute, but with gradually increasing gate opening, the speed and load being practically constant. This was due to the water rising in the forebay, and gradually subsiding in a succession of waves, the governor taking care of these fluctuations, in effective head, in a very intelligent manner." *

"The plant in which these tests were made was by no means a good one from the regulation standpoint, for it will be noticed that when the whole load was instantly thrown off the momentary rise of speed was about eight per cent., although the governor shut the gate from full open position in the extremely quick time of one and four-tenths seconds. There were five wicket gates, having a total of ninety-six leaves, and a heavy counter-weight to be moved a considerable distance in this interval.†

224. General Consideration.—Mechanical governors are cheaper than hydraulic, but, assuming the same gate movement, they are less effective at increasing loads since the power to move the gates must be taken as needed from the wheel itself instead of being taken from a storage tank as with hydraulic governors. This is a factor of more or less importance in accordance with the degree of regulation required. The difference is manifest principally at low loads when the energy taken by the governor relay from the water wheel is a considerable percentage of the total energy being generated. As the power exerted by the relay is usually comparatively small, the difference in action from this cause between the two types of governors is often unimportant.

The hydraulic governor possesses an additional advantage in its ability to start a stationary wheel into action by means of its stored energy. The mechanical governor, depending as it does on the power of the wheel itself, is only effective after the wheel has been started by other means.

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* See American Society M. E., Vol. 27, No. 4, p. 8.

† Catalogue of Water Wheel Governors, Sturgess Engineering Department of the Ludlow Valve Co., p. 23.

225. The Glocker-White Turbine Governor.—The I. P. Morris Company has built a governor for the Electrical Development Company of Ontario, Canada, which has one novel feature.* A cross-section of its distinctive feature is shown in Fig. 302, page 481.

The governor ball is hollow and contains two chambers, *a* and *b*, communicating with each other through a small opening, *c*.

The balls are partially filled with mercury which, when running at normal speed, the axis of the ball being vertical, is divided between the two chambers. When an increase of speed throws the balls outward, centrifugal force causes a flow of mercury from chamber *a*, to chamber *b*. This raises the center of gravity of the ball and increases its lever-arm about the knife edge *j*, thus increasing its effectiveness by making its movement increase in a greater ratio than the speed increases. Similarly a reduction in speed causes the balls to incline inward and the mercury therefore to flow from chamber *b*, to chamber *a*, which tends to cause a still greater inward inclination.

The charge of mercury hence increases the sensitiveness of the governor balls to small changes in speed.

The centrifugal force of the balls is resisted through knife edges *K, K*, by a spiral spring. This movement is transmitted by levers to a small pilot valve which controls a larger relay valve admitting oil under 250 pounds pressure to the cylinder. The gate to be moved is a cylinder gate opening upward, a force of 15,000 pounds being required for the purpose. The weight of the gate is sufficient to close it and the power-cylinder of the governor is therefore made single acting. The entire governor is not shown as there are no other unusual features.

226. The Allis-Chalmers Governor.—This company has recently developed a water wheel governor, the following description of which is taken from their bulletin No. 1612:

"The Allis-Chalmers Governor is of the oil pressure type and consists of three distinct elements:

"First—Governor Stand (see Fig. 303, page 482) containing the apparatus for spindle, *f*, to rise and fall changes the position of the tappet arm, controlling the time of application of energy for actuating the gates.

"Second—Regulating Cylinder for applying energy.

"Third—Pressure System for supplying energy.

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* See "The Glocker-White Turbine Governor" by W. M. White and L. F. Moody in "Power," Aug. 4, 1908.

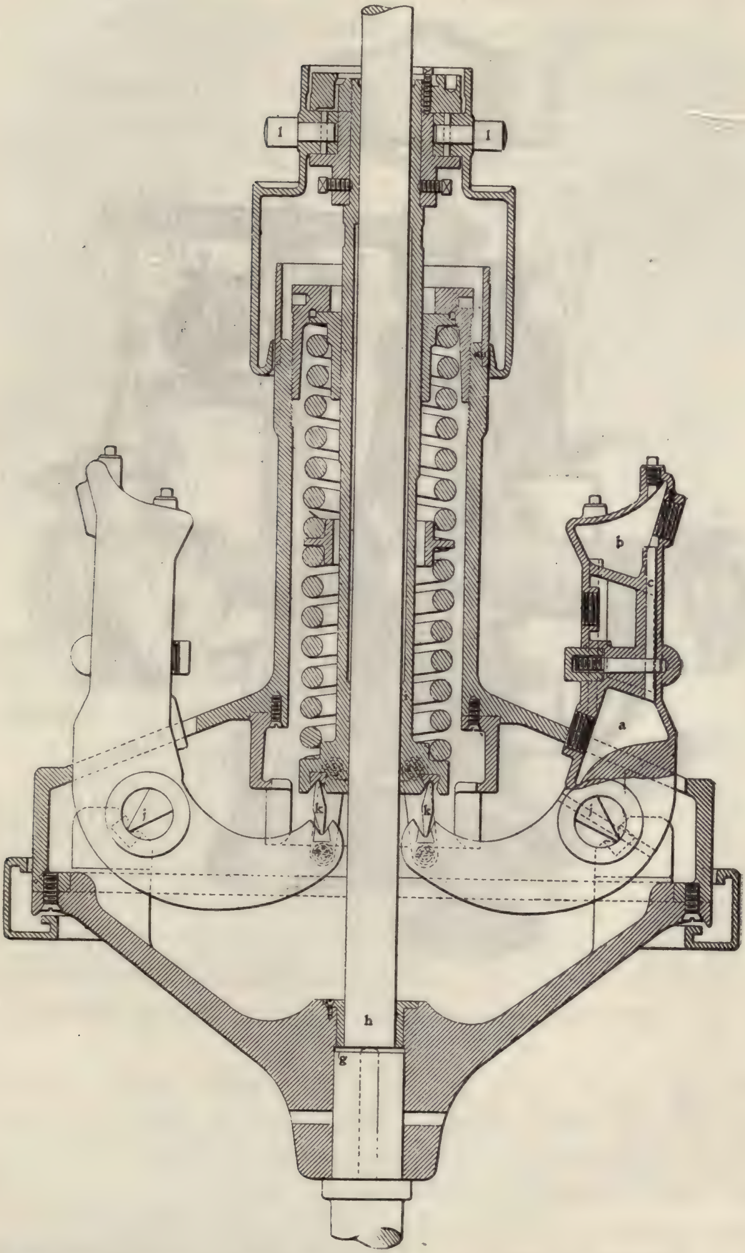

Fig. 302.—Cross-Section of the Glocker-White Governor Head (see page 480).

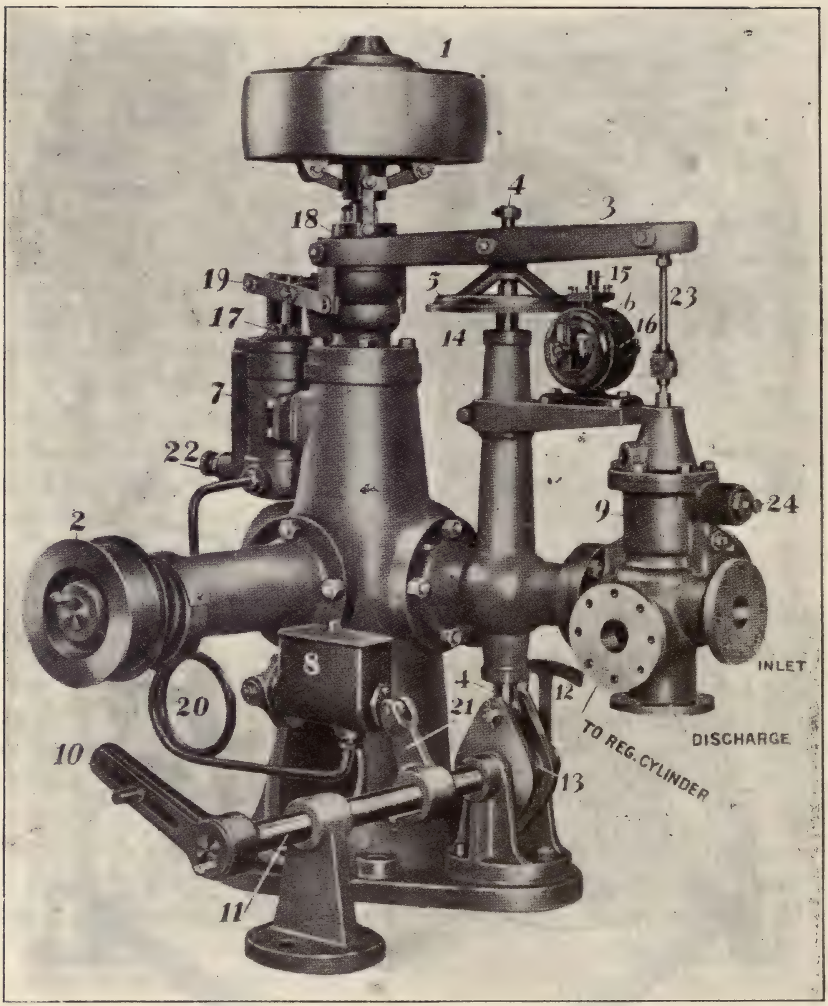

Fig. 303.—View of the Governor Stand of the Allis-Chalmers Governor (see page 480).

"The governor head 1, designed to be a highly sensitive yet stable apparatus and driven from the turbine shaft by pulley 2, forms the basic governing element. Any change in its position moves the governor collar 18, thereby shifting the floating lever 3, and through it and its connection with the relay 4 (which momentarily acts as a stationary fulcrum) actuates the regulating valve 9. Any movement of this regulating valve admits oil from the pressure system to either the

opening or closing side of the regulating cylinder and thereby actuates the turbine gates. The relay 4, forms a mechanical connection between the regulating cylinder piston and the floating level 3, constituting what may be termed a moving fulcrum, so that every movement of the regulating piston shifts the fulcrum point and brings the regulating valve 9, back to mid position, thereby making the mechanism 'dead beat.' If this movement is adjusted so that the position of these parts have the proper relation, the governor collar will practically retain a fixed position.

"The regulating cylinder cannot, however, fully open or close the turbine gates instantaneously and the above result can only be obtained within certain limits, a difference of speed occurring between no load and full load that requires a certain movement or travel of the governor collar 18. Consequently, the speed of the turbine at different gate

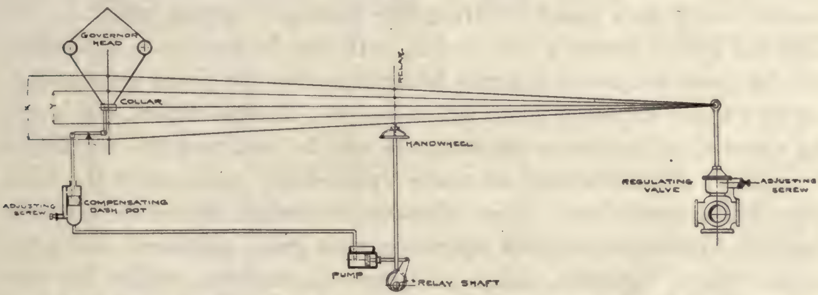

Fig. 304.—Diagram of Allis-Chalmers Governor (see page 480).

openings will vary slightly and depend upon the speed of the governor at corresponding positions of the regulating piston stroke.

"Under favorable conditions (open flume and short penstocks) the opening and closing time of the gates depends solely upon the inertia of the moving masses and 'aperiodical regulation' can be obtained; i. e., the stroke of the regulating piston and the travel of the governor collar correspond in time. Under favorable conditions (long penstocks) the closing time is often so influenced by the 'critical time,' already mentioned, and by other considerations that 'aperiodical regulation' is no longer practicable, since a travel of governor collar would be required that would cause a greater difference in speed between no load and full load than is commercially allowable. To meet such conditions, the "compensating dash pot" 7, is utilized.

"In the diagram (see Figure 304), the full travel of the governor collar is shown as corresponding to a speed change x. The relay

stroke, however is designed so that only a portion of this travel corresponding to a speed change y is utilized; i. e., within this limit the governor, without other mechanism than the relay, is 'dead beat' and the regulating valve by relay action is returned to mid-position after each movement. The compensating dash pot 7, consists of a cylinder having an adjustable bypass and containing a compound piston with auxiliary spring device, the rod of which is connected through a suitable lever to the governor collar. Arranged so that its piston takes motion from the relay actuating shaft, is a positive displacement pump connected by a pipe to the 'compensating dash pot' cylinder. For slight changes of load, a negligible displacement of oil takes place and the dash pot has a slight damping action only on the governor head, but when any load change occurs of sufficient magnitude to produce a speed variation greater than y as shown on the diagram, enough oil displacement takes place to bring the auxiliary spring effect of the dash pot piston strongly into action until the fluctuation is controlled and the governor collar is again brought within the limits corresponding to y speed variation when action ceases. By this means, a governing element of maximum sensitiveness can be used and the regulation of ordinary slight fluctuations made 'aperiodical,' even under the most unfavorable conditions. These elements in design, therefore, result in the Allis-Chalmers governor operating with great quickness and holding the speed variation, due to ordinary fluctuations, within the narrowest limits, yet being absolutely safe from hunting or over-travel after heavy load fluctuation, even under the most difficult operating conditions."

227. **Control from the Switchboard.**—Electrical devices can now be purchased by which the normal speed of the wheels can be controlled from the switchboard in case the governor is so designed that it can be adjusted while in motion, which is true of most high class machines. It is also possible to start and stop the wheels electrically from the switchboard or from a distant station.

The following discussion of this subject and the accompanying figures are taken with slight changes, from a paper by Mr. A. V. Garratt.*

" * * * The most successful method of connecting the cylinder gates of several turbines to the same governor is shown in Fig. 305, page 485. In this case each pair of drawrods is connected to a pair of

* See "Speed Regulation of Water Power Plants," by Allan V. Garratt. Cassier's Magazine, May, 1901.

walking beams which carry counterweights on their opposite ends. Each walking beam carries a gear sector which engages a rack on a long, horizontal reciprocating member terminating at the governor. The racks on the reciprocating member are 'sleeved' on it, and held in place by pins, which may be removed if it is desired to disconnect any turbine from the governor.

"By this method any one, or any combination of turbines, may be handled by the governor or any turbines by hand, at will, by means of a lever shown in the end projection.

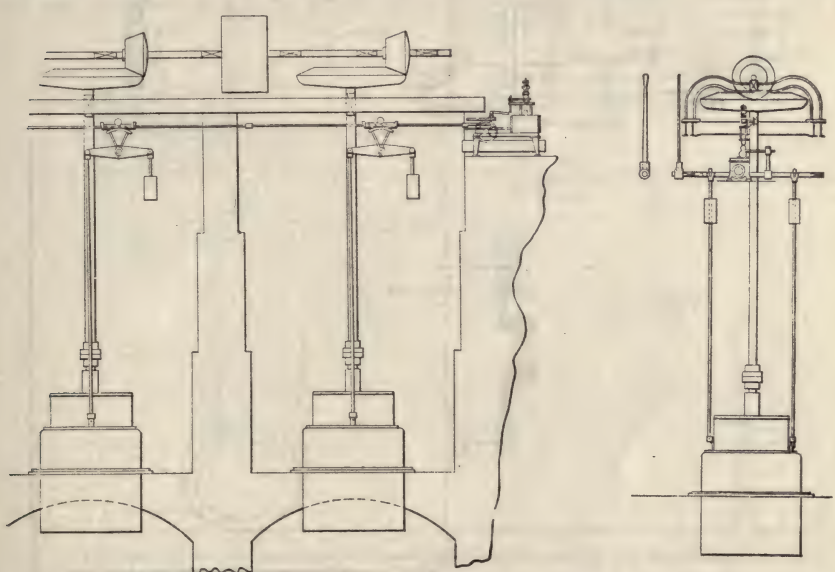

Fig. 305.—Governor Connection by Draw Rods (see page 484).

"Fig. 306, page 486, shows a good method of connecting a governor to a pair of horizontal wicket-gate turbines. It will be noted that the shaft connecting the two gear sectors on the gate stems goes directly to the governor, and is connected to it through a pin clutch which may be opened, and a hand-wheel on the governor may then be used to move the gates by hand. The only improvement on this design which can be suggested would be to eliminate the counter-shaft between the governor pulleys and the turbine shaft by placing the governor beyond the draught-tube quarter-turn, so that the governor pulleys might belt directly to the turbine shaft. The limitations of available space prevented the location of the governor in this manner on the drawing which shows the design used for three units in a modern power plant.

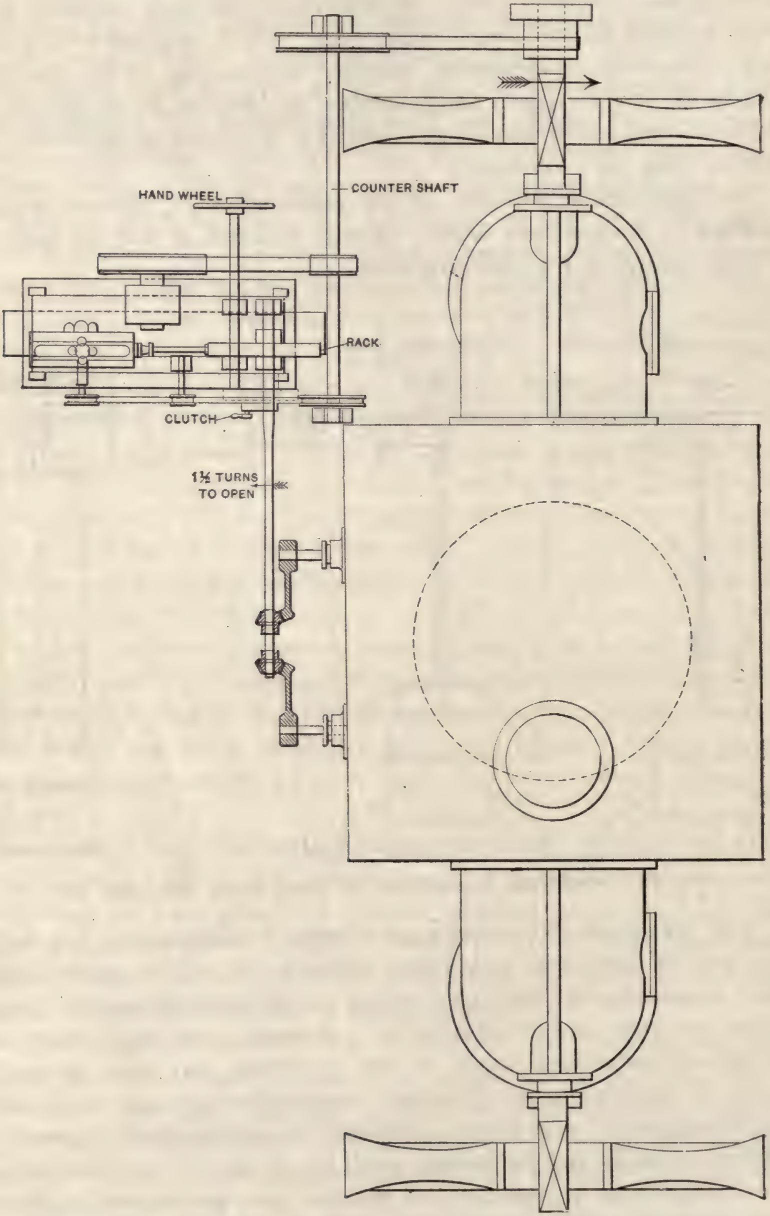

Fig. 306.—Governor Connection by Shaft and Sectors (see page 485).

"Frequently the only possible location of the governor prevents anything like direct connections between it and the turbines. In such cases experience has shown that it is wisest to avoid the use of several pairs of bevel gears and long shafts, and in their place use a steel rope drive. This method has great flexibility, and permits of governor locations which would otherwise be impossible. Figure 307 shows a design of this kind. The governor is located in the only available space, and yet its connection to the turbines is perfectly adequate. The steel rope used is small in size, made of very small wire, especially laid up, and its ends are fixed to the grooved sheaves, which are provided

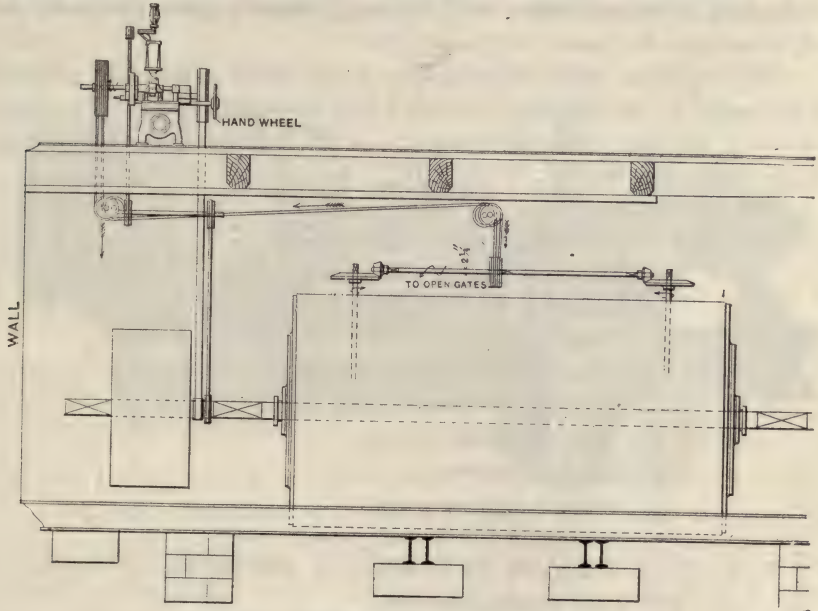

Fig. 307.—Governor Connection by Cable.

with internal take-ups, so that the rope may be kept tight as a fiddle string. This general method of connecting is in successful use in many plants where the requirements for speed regulation are most exacting.

"In the above examples the two ends which have governed the design are simplicity and directness. These two factors should never be lost sight of, and the more completely they are embodied in the design, the better will be the speed regulation. To these two may be added another, and that is freedom from lost motion. These three factors are absolutely necessary if successful results are to be expected. The slightest motion of the governor must be transmitted in the simplest

and most direct manner, and in the shortest possible interval of time, to the turbine gates."

228. Governor Systems.—For the governing of hydro-electric plants, the hydraulic governors used are of two distinct classes:

First: The open system, in which the oil used for the transmission of pressure from the pumps to the actuating cylinders of the governors is discharged from those cylinders at atmospheric pressure into open tanks, and

Second: The closed system in which the oil in the circuit is closed to the atmosphere in order that the discharge end of the actuating cylinders of the governors may discharge under a partial vacuum and thus increase the power of the governor.

These systems were discussed in some detail before the thirtieth convention of the National Electric Light association at San Francisco in 1915, and the following summary of the advantages and disadvantages was given by Mr. W. F. Uhl:

The advantages are:

(a) For the closed system—

(1) The receiving or sump tank can be placed at about the same level with the regulating valve.

(2) The lower first cost for small turbines.

(b) For the open system—

(1) Simpler equipment.

(2) Less operating trouble.

(3) Smaller maintenance cost.

(4) Smaller operating cost.

(5) Adaptable for central system use.

(6) Adaptable for straining or filtering the oil constantly.

(7) Less power required.

The disadvantages are:

(a) For the closed system—

(1) Operating trouble, principally on account of the rapid deterioration of the oil.

(2) When changing from mechanical hand control to governor control after the pumping equipment has been running and building up a vacuum, the gates may suddenly close, causing a high pressure rise, which in certain cases might be destructive.

(3) Closer attendance required.

(4) Greater operating cost.

(5) Not adaptable for central system.

(6) Not readily adaptable for straining or filtering the oil.

(7) More power required.

(b) For the open system—

(1) Not easily adapted to certain low head plants having horizontal units on account of the location of the receiving or sump tank being necessarily lower than the regulating valve. Not many such cases with modern installations.

This discussion, which can be found in the reports of the Association for 1915, contains the most practical information available on this subject.

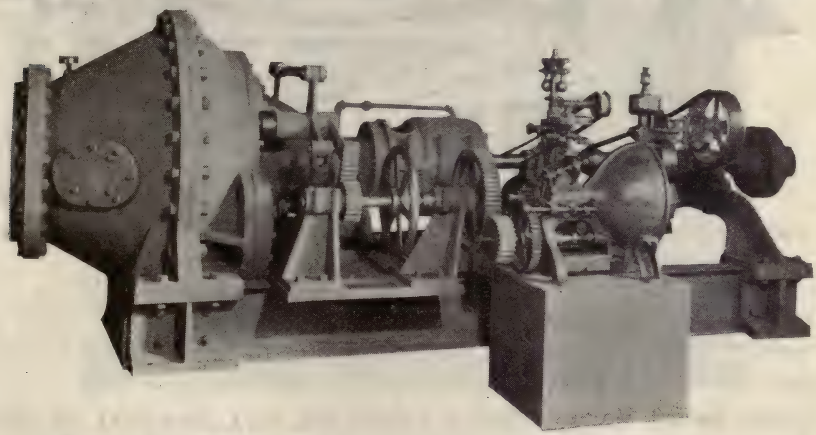

Fig. 308.—Connection of a Modern Governor to a Victor Turbine.

229. The Connection of Governors to Gates.—The governor should be connected to the gate mechanism of the turbine as directly and substantially as practicable. The connection of a modern governor with a Victor turbine (Platt Iron Works) is shown in Fig. 308.

Figure 309, page 490, shows the connection of an Allis-Chalmers governor with both the gates of a single horizontal turbine operating under 450 feet head and with the relief valve, which must likewise be controlled during sudden reductions in power demand.

Figure 310, page 490, shows the connecting mechanism for a twin turbine unit of 5,200 horse power operating at 225 R. P. M. under 60 ft. head, as built by the S. Morgan Smith Company.

The form of connection for a six turbine unit, as manufactured by the Wellman, Seaver, Morgan Company, is shown in Fig. 340, page 528,

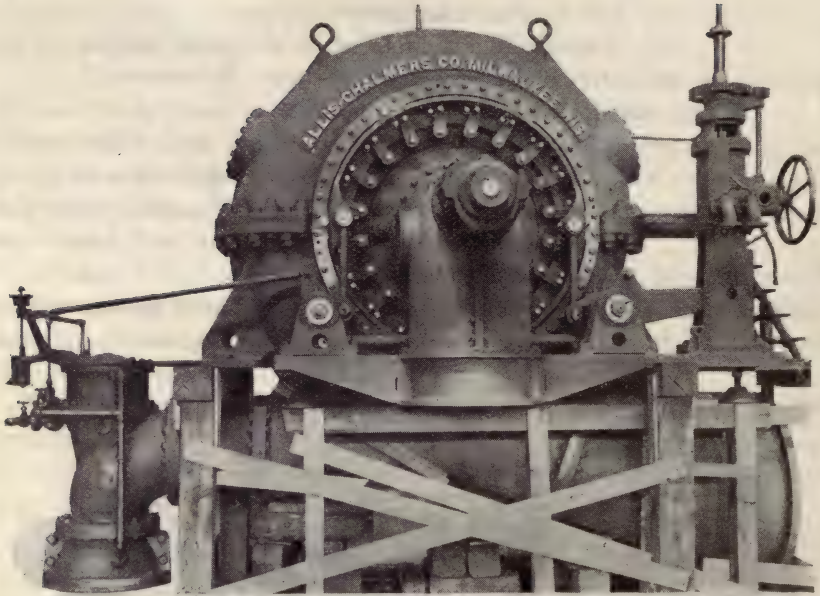

Fig. 309.—Connection of an Allis-Chalmers Governor with the Gates of a Horizontal Turbine and with a Relief Valve (see page 489).

and Fig. 341, page 529, shows a similar connection for four pairs of turbines arranged tandem as manufactured by the James Leffel & Company.

230. Relief Valves.—Relief valves are very necessary on long feeder pipes and penstocks to avoid excess pressures of an accidental nature as well as those produced by closing of the turbine gates. A

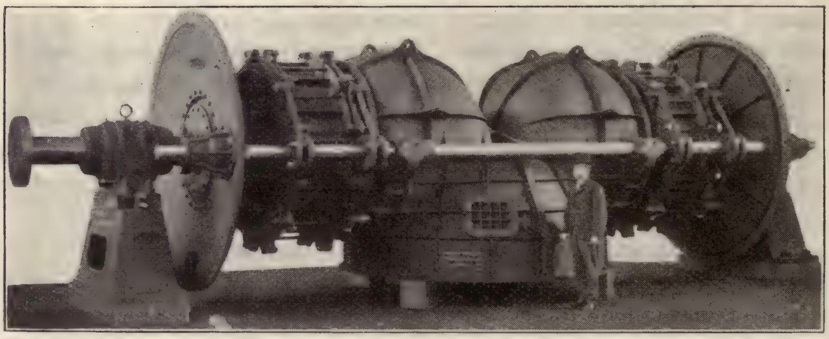

Fig. 310.—Governor Connecting Mechanism for a Twin Turbine of the S. Morgan Smith Co. (see page 489).

group of such valves installed on the end of one of the penstocks of the Niagara Falls Hydraulic Power and Manufacturing Company is shown in Figure 311. Relief valves should be arranged to open with a slight excess of the penstock pressure but should close very slowly in order to avoid oscillatory waves. Spring balanced relief valves have proven objectionable for this purpose. If set to open at a small excess pressure they are apt not to close on account of the impact of the discharging water against the valve. In order that they may

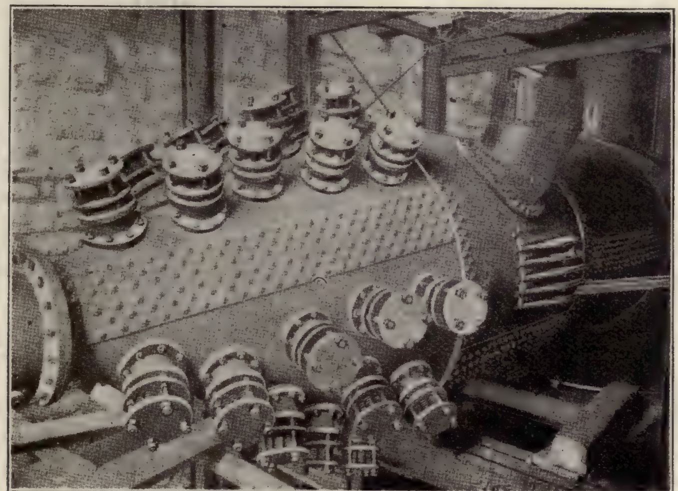

Fig. 311.—Relief Valve on end of Penstock. Niagara Falls Hydraulic Power Manufacturing Co. (Electrical World, Jan. 14, 1899).

close, the balancing spring must be so strong that a considerable excess is required to open the valve which does not therefore serve the desired purpose. All types of valves are also hindered by the fact that corrosion is apt to seal the valve so that a considerable excess is required to open it.

231. **Lombard Hydraulic Relief Valve.**—The Lombard Governor Company have designed a valve in which they claim to have eliminated the difficulties of the spring valve. This valve is shown in Fig. 312,* page 492, and is described as follows:

"The valve consists of the following parts, viz.:—A valve disc *c*, capable of motion to or from its seat *b*, rigidly connected by means of a rod *i*, with the piston *f*, in the cylinder *e*. The whole valve is bolted

* Lombard Bulletin No. 101.

to a flange upon the supply pipe *d*, wherein the pressure is to be controlled. The area of piston *f*, is somewhat greater than that of the valve disc *c*, so that when water at the same pressure is behind the piston and in front of the valve there is a positive and strong tendency

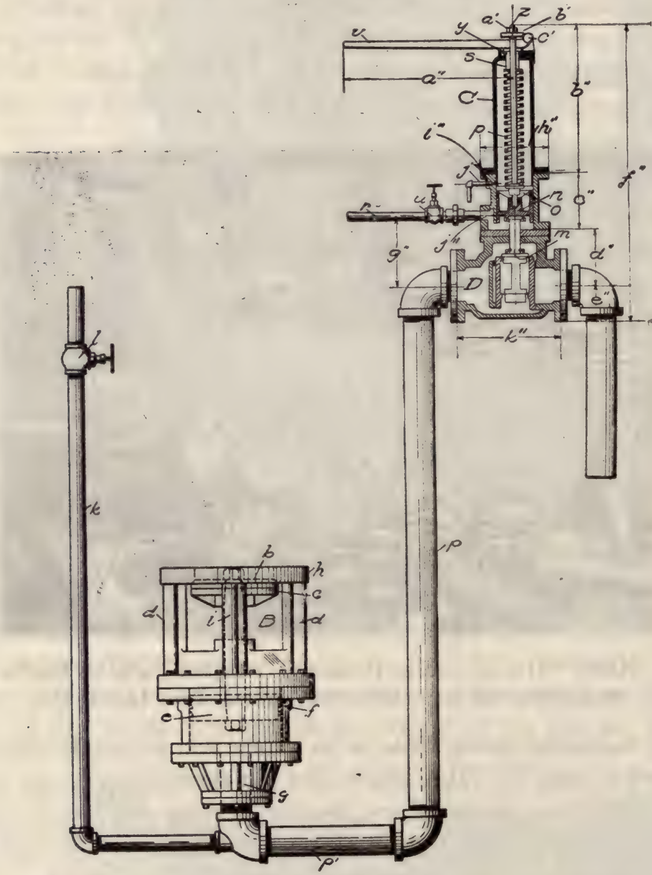

Fig. 312.—Lombard Hydraulic Relief Valve (see page 491).

to hold the valve closed. For the purpose of allowing the valve disc *c*, to open at proper times to relieve excess pressure in the supply pipe *d*, there is provided a regulating waste valve *C*. This valve is opened or closed by a piston *n*, opposed by a very oblong and strong spiral spring *p*. Piston *n* is a loose fit in its cylinder *o*, so that it moves upward freely in response to the least excess in pressure upward due to the water in the cylinder *o*, opposed to the downward pressure of the

spring *p*. * * * The piston *n*, is connected by means of the stem *m*, with a double-seated balanced valve *d*, which of course, opens simultaneously with any upward movement of the piston. Water under existing pressure is admitted into the cylinder *e*, through the pipe *k*, and throttle valve 1.

"The spring *p*, is adjusted by means of the screw *s*, and lock-nut *y*, so that the effective normal pressure of the water in the chamber is just insufficient to overcome the downward pressure of the spring. The valve *D*, will therefore remain closed normally; consequently the main valve disc *c*, will also remain closed normally, because water flowing in through the pipe *k*, and throttle valve 1, will produce an excess closing pressure upon the piston *f*. When thus adjusted any increase in pressure above the normal will immediately force the piston *n*, upward, and will thereby open the balanced valve *D*. This instantly relieves the pressure back of the piston *f*, which of course then gives way to the superior pressure back of the piston *f*, which of course then gives way to the superior pressure in front of valve *c*. In this manner practically the whole pressure in front of the valve disc *c*, is available for opening it. * * * Valve disc *c*, will continue to open until the limit of its travel has been reached, or the pressure in the supply pipe *d*, has been reduced to a point where the piston *n*, will close the balanced valve *D*. Immediately on the closing of balanced valve *D*, water begins to accumulate behind the piston *f*, flowing in through the throttle valve 1. This water gradually and surely forces the valve disc *c*, to close. The speed of closing is adjustable by the opening through the throttle valve 1, and may be made as slow as several seconds or even minutes. The closing motion is * * * uniform and there is not the slightest tendency to set up vibrations in the water column, a very serious objection to the ordinary types of spring balanced valves which open and close suddenly and are liable in the latter operation to set up water hammer effects even more dangerous than those which they are designed to relieve."

232. **Sturgess Relief Valves.**—The Sturgess Engineering Department of the Ludlow Valve Manufacturing Company makes two forms or relief valves, the "automatic" and the "mechanical." The automatic relief valve is shown in Fig. 313, page 494, and is described as follows:

"The essential element in the automatic relief valves is a large, very sensitive diaphragm of special construction. This is under the influ-

ence of the water pressure in the pipe-line and its movements are communicated to a small pilot valve controlling a hydraulic cylinder, which in turn operates the relieving valve on the relief valve proper. After

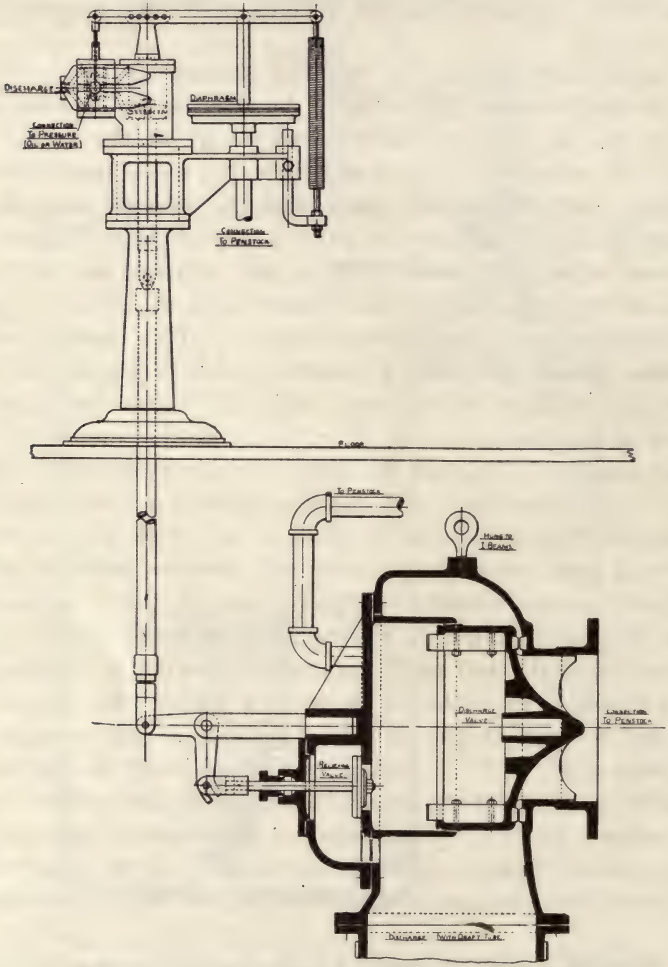

Fig. 313.—Sturgess Relief Valve (see page 493).

the pressure in the pipe-line is restored to normal, the relief valve gradually closes automatically.

"The action of this valve is almost instantaneous, and it will fully open on a very small rise of pressure.

"These valves can either be made in self-contained form, or the sensitive parts (diaphragm, pilot valve, and hydraulic cylinder) may be

mounted on a pedestal placed in the power house, and the relief valve proper attached to the penstock or wheel casing, a rod or link being provided to connect the two (as in Fig. 313).

233. Allis-Chalmers Company Hydraulic Relief Valve.—Figure 314 shows an automatic governor actuated pressure regulator for 7,000 H. P. Allis-Chalmers Company turbines, operating under a 450 foot head with a discharge capacity of 210 cubic feet per second.*

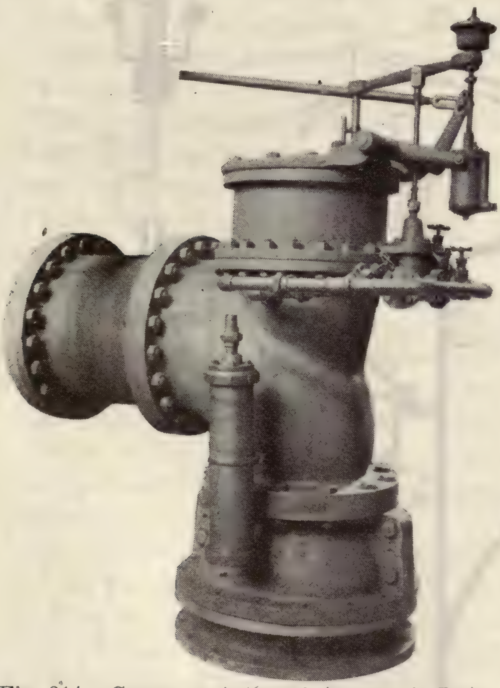

Fig. 314.—Governor Actuated Automatic Pressure Regulator for 7,000 H. P. Turbine.

"Each pressure regulator consists of two distinct elements, namely: the by-pass proper and the regulating valve operated from the relay mechanism.

"The various parts and the operating function are made plain from the following description, and reference to Fig. 315, page 496:

"The by-pass consists of the inlet elbow 1, connected to the casing or inlet pipe of the turbine, and the discharge ring 2, bolted to the foundation of discharge pipe. The bronze lined valve stem 3, carries the valve disc 4, and the piston 5, sliding in the bronze bushed cylinder 6, with cover 7, forming the upper extension of the inlet elbow 1. A removable bronze ring 8, is clamped between discharge ring 2, and inlet elbow 1, and forms the seat against which the valve disc 4, is pressed against its seat when the pressure regulator is closed. The valve disc 4, is also lined with a removable bronze ring 23. Parts 8 and 23, being subject to wear and tear due to the discharging water, are made removable and interchangeable.

"The piston 5, has a *V* leather packing and its area is sufficient to hold the valve disc 4, tightly closed when the cylinder *A*, is set under

* Allis-Chalmers Co.

pipe line pressure. The water which might leak through the *V* leathers is drained into discharge by the pipe tapped into the cover 7.

"If the pressure is reduced in the cylinder *A*, the pipe line pressure upon the valve disc 4, will open the pressure regulator and it is evi-

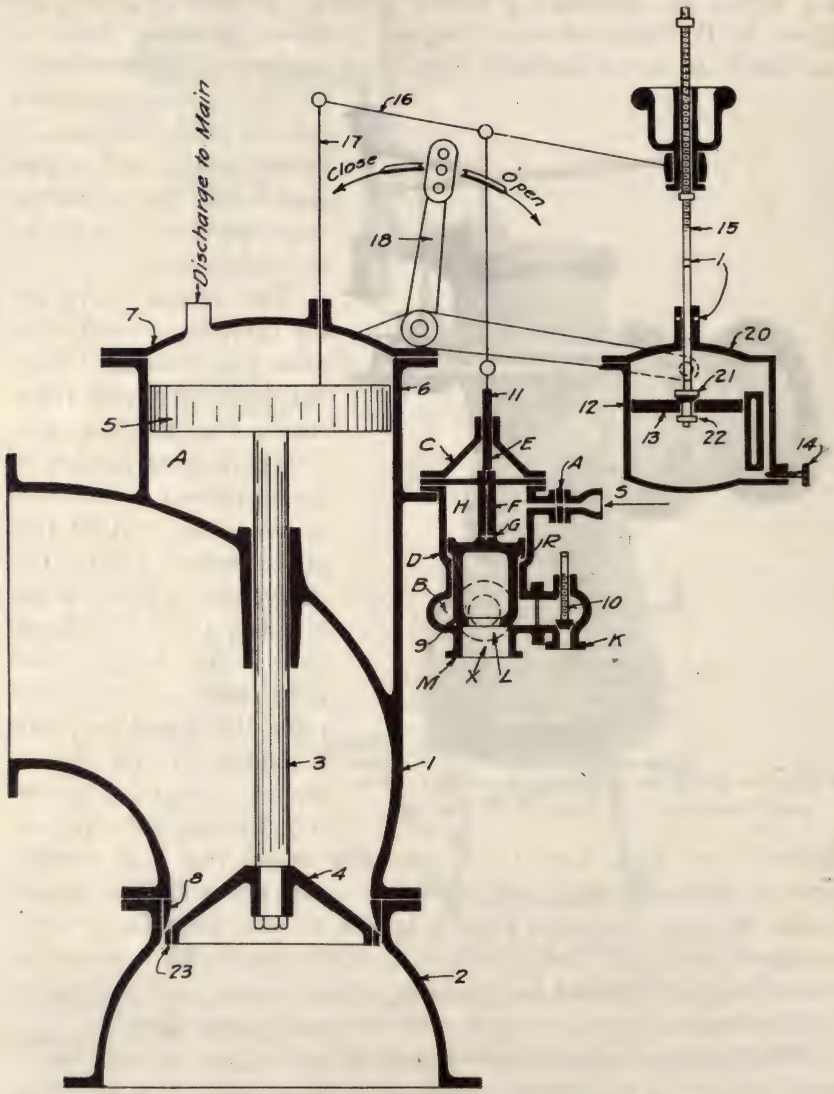

Fig. 315.—Cross-section of Allis-Chalmers Co. Governor Actuated Pressure Regulator (see page 495).

dent that if we vary the pressure in the cylinder *A,* we can operate the valve disc 4, in either direction with any desired velocity.

"This is accomplished by the second element, the regulating valve and relay mechanism. The regulating valve is bolted to the regulator cylinder 6. It has three flanges, *K, L, M.* Flange *K* is connected to the pipe line and receives direct operating water pressure. (A filter should be inserted in this pipe in case the water in pipe line is impure.)

"Flange *L,* connects chamber *B,* with pipe leading to the cylinder, *A.*

"Flange *M,* connects to the discharge pipe.

"*C* connects to the drain.

"The area of pressure inlet into chamber *B,* can be adjusted from outside by the needle valve 10, and the outlet from chamber *B,* is regulated by the valve 9 actuated by the stem 11, as explained below.

"Stem 11, is attached to floating lever 16, one end of which is connected to relay rod 17, while the other end is attached to dash pot piston rod 15.

"Lever 18, is connected to the gate mechanism of the turbine and also to the dash pot cylinder 12, which contains the dash pot piston 13, loosely coupled to piston rod 15, by means of taper collar 21, and collar 22. Needle valve 14 allows the by-pass adjustment from outside.

"Water under penstock pressure is introduced through diaphragm *A,* into chamber *H,* also through opening *D,* without diaphragm into chamber *R.*

"At a given instant when gate and governor mechanism are at rest, let *P* be the total pressure in chamber *H,* holding valve 9, closed and let P^1 be the total pressure in chamber *R,* tending to open valve 9. From the respective areas evidently *P* is greater than P^1, hence valve 9, remains closed.

"Now suppose the governor to close the gates considerably, thus raising lever 18, and with it the valve stem 11. This opens the valve *E,* allowing some of the pressure in *H* to escape through orifices *G, F* and *E,* until the variable pressure *P,* is overcome by P^1, causing valve 9, to rise, allowing the water pressure in cylinder *A,* to escape through *B* and opening *X* to the drain. The lowering of pressure in *A* permits the pressure regulator to open under the penstock pressure acting on valve disc 4. It is evident that a certain position of valve 9, permits a certain pressure in chamber *B,* sufficient to hold the piston 5, and consequently the pressure regulator, in a balanced intermediate position. When valve *E,* is closed, pressure *P,* increases so as to close valve 9 causing the pressure in *B* and *A* to increase and close the

pressure regulator valve 4. in a time which depends upon the setting of needle valve 10.

"When the turbine closes, the mechanism is arranged so that the lever 18, raises the dash pot cylinder 12. If this motion is so slow that the oil below the dash pot piston 13, can escape to opposite side of piston through the by-pass opening set by the needle valve 14, there will be no over-pressure produced below the dash pot piston; the valve stem 11, will not be raised from its seat *E,* as described above and the pressure regulator remains closed. If, however, the turbine is closed rapidly, i. e., in a time shorter than to allow the oil getting around dash pot piston 13, the thus created over-pressure will raise valve stem 11, and cause the pressure regulator to open. As soon as this takes place the relay rod 17, is lowered accordingly and thus closes the valve stem 11, and brings valve 9, into the position at which the pressure regulator is held stationary. If the lever 18, is not moved any farther the dash pot piston 13, on account of the gradual by-passing of the oil through valve 14, will slowly move downward and thus slowly close the pressure regulator. It is evident that the closing time of valve 11, depends entirely upon the setting of the needle valve 14. This setting cannot be given in advance but must be tried out by experiment at the plant. It should be such that the pressure regulator closes sufficiently slowly to prevent any pressure rise in pipe line. Long pipe lines or such through which the water flows at high velocity, will require a long closing time, sometimes as long as 50 seconds."

234. Speed Regulation.—Mr. H. B. Taylor, hydraulic engineer of the I. P. Morris Company in a paper read before the Canadian Society of Engineers, January, 1914, discussed the matter of speed regulation as follows:

"Speed regulation is of great importance where turbines are employed for driving alternators. Regulation is obtained by a governor, driven from the turbine shaft, consisting primarily of a governor head containing flyballs controlled by springs and connected through a pilot valve to the main governor valve. At normal speed the flyballs hold the governor valve in mid-position, but for any change in speed the flyballs move the valve, admitting pressure through a suitable number of relay valves to an operating cylinder so that the turbine gates are adjusted to suit the new load and to obtain the normal speed.

"Some of the chief speed-regulation requirements are:

"(1) Sensitiveness—quick response to sudden and slight changes of speed.

"(2) Steadiness—freedom from unnecessary gate movements or movements occasioned by conditions other than changes of load. Continual motion and vibration of the turbine gate mechanism produced by unstable governors is a source of wear and rapid deterioration which in turn accentuates the instability. In older types of governors this tendency was usually overcome by the use of a dash pot to damp such motions, but the use of a dash pot for this purpose renders the governor sluggish and impairs its action.

"(3) Ample power of governor. For large units a large amount of power is required to actuate the turbine gates. With a large unit, either a considerable number of intermediate valves between the pilot valve and the main valve must be used, or else the governor head itself must be given a considerable amount of power. The same result could perhaps be accomplished by the use of high pressures in the governor system, but these have proved very unsatisfactory. Increase of power by a series of valves is limited by the speed of action required. The best solution seems to be in the use of flyballs, having a weight and power corresponding in order of magnitude to the turbine which they must control. Such increase in power in the flyballs involves no sacrifice in sensitiveness, since the governor head is mechanically connected to the turbine shaft and is forced to respond immediately to any change in speed of the unit. The sensitiveness will, in fact, be increased by the use of heavy flyballs, since the retarding effect of friction can be made relatively less.

"(4) Reliability. Delicate and complicated mechanisms should be avoided. Governors in important stations should be as simple and rugged as possible. The effect of accidental conditions on a large and heavy governor, grit in the governor fluid or sticking of the governor parts, will be comparatively slight.

"(5) Single-pressure system. Separate pumping equipment for each unit in a station has been abandoned in favor of a single pumping system supplying the entire station with greatly reduced cost of attendance and greater continuity of service. The open system of governing is now being used in which oil or water for the governors is pumped from an open tank and no pressures less than atmospheric are used at any point in the system. This change avoids troubles caused by air collecting in the pipes or pumps or the breakdown of oil under high vacuum.

"(6) Good hand control. Originally the hand control of a turbine was through trains of gears. The time required to close the gates of

a large turbine by such means is prohibitive. This gear was gradually replaced by a governor hand wheel controlling the turbine gates by oil pressure through the governor main valve. However, should the governor valves be dismantled for repairs the turbine must be shut down, and it is poor engineering to dispense with the services of the turbine for the sake of a governor valve. The best method of hand control now adopted for large units, seems to be in a separate device to admit oil from the central pumping system directly to the operating engine.

"(7) Few simple operations. Compensating devices are required to restore the governor valves after required gate movement and also to restore the speed of the turbine to required value after the water column in penstock, turbine casing and draft tube has had time to be accelerated or retarded to its new value. The governor should also have adjustments, so that the change in speed from full load to no load conditions may be fixed to suit the parallel operation of alternators, also to adjust the time of action of the governor to suit the length of and velocities in the water passages, and an attachment for changing the normal speed of the turbine. Modern governors are also fitted with motors by means of which the speed of the turbine can be adjusted from a distant point, such as the switchboard, this device being used in synchronizing the unit.

"The above points cover all the features which can be controlled in the design of the governor itself.

"A perfect governor will be unable to produce a speed regulation better than that permitted by the length of and velocities in the penstock and draft tube, the length of the water column in the turbine casing and the flywheel effect of the rotating masses. These factors are controlled by the design of the power development as a whole, so that the actual speed regulation obtained is affected only to a limited extent by the construction of the governor itself.

"In a majority of the recent large installations the governors have been designed and constructed by the turbine builders suited to the exact conditions of each turbine and avoiding division of responsibility. The better appreciation of the factors entering into governor problems has resulted in greatly improved speed regulation which, when taken in connection with valuable improvements in the mechanical design of the governor, has removed this important auxiliary from the class of necessary evils and placed it in that of reliable machinery."

235. Governor Specifications.—The present practice of requiring the governor builder to guarantee the speed regulation of a plant, in the design of which he has had no voice, without even giving him the necessary information regarding the hydraulic elements which are considered in this chapter is wrong. It is partly the outgrowth of the modern tendency to specialize, but perhaps more largely due to a lack of understanding on the part of the engineer of the nature of the problem, and a resulting desire to shift the responsibility for results upon some one else who is better informed upon the subject and thus protect results financially as well as save his own reputation in case of failure.

Governor specifications should call for a guarantee of the

(a) *Sensitiveness* or per cent. load change which will actuate the governor;

(b) *Power* which the governor can develop, and *force* which it can exert to move the gates;

(c) *Rapidity* with which it will move the gates;

(d) *Anti-racing qualities*, such as number of gate movements required to adjust for a given load change (see Fig. 280, page 429), or per cent. over-run of the gate, etc.

(e) *General requirements* of material, strength, durability, etc.

Beyond this point the governor designer has no control. The engineer can, however, by choosing a generator whose rotor has a high moment of inertia (which quantity should be stated in tenders for supplying the generators), by the addition of a fly-wheel, if necessary (see Fig. 333, page 523); by the construction of a stand-pipe; by means of a relief valve, and very largely, also, by the general design of the penstocks, draft tubes, etc., greatly improve the governing qualities, and, in fact, reduce the speed variation to any desirable limit which the nature of load to be carried, magnitude of load changes anticipated, and economy of first cost will warrant.

CHAPTER XVI

ARRANGEMENT OF THE REACTION WHEEL

236. General Conditions.—The reaction turbine may be set or arranged for service in a water power plant in a variety of ways, and the best way may differ more or less with each installation. The arrangement of wheels should always be made with due regard to machinery to be operated, the local conditions that prevail, and especial consideration should be given to securing the greatest economy in the first cost of installation, maximum efficiency and facility in operation, and minimum cost of operation and maintenance.

Impulse water wheels of the tangential type have always been set with their shafts horizontal. An installation with vertical shaft was proposed for one of the first Niagara plants but was not considered on account of the lack of actual experience with such a form of installation. Impulse wheels of the Girard type have been used with both vertical and horizontal shafts. In general, however, because of the high heads under which impulse wheels usually operate, the horizontal shaft arrangement is readily adapted. When an impulse wheel is installed it must be set above the level of maximum tail water, if it is to be operated at all stages of water. The wheel arrangement is therefore dependent principally on the arrangement of the machinery to be operated. By far the greater proportion of such machinery is built with horizontal shafts and hence in most cases where machinery is not special, horizontal shaft arrangements are desirable.

Reaction wheels are often used on streams where the relative variation in position of the tail-water is considerable, and it is both desirable to utilize the full head and to have the wheel set at an elevation at least above the lowest elevation of the tail-water in order that they may be accessible for examination and repairs. By the use of the draft tube this can often be done without the sacrifice of head. If the wheel must be set below tail-water, gates must be provided for the tail-race with pumps for the removal of the water when access to the wheels is necessary.

The arrangement of reaction water wheels is susceptible only of general classification, which, however, may assist in the understanding of the subject and the selection of the best methods to be adopted un-

der any set of local conditions. Wheels may be set vertically or horizontally, as the conditions of operation demand, without materially affecting their effciency, provided that in each instance the turbine case, draft tubes, etc., are suitably arranged. The improper design of the setting may materially affect the efficiency of operation in either case.

237. Necessary Submergence of Reaction Wheels.—In order to prevent the formation of a vortex or whirlpool, which will draw air into the wheel and often seriously affect its power and efficiency, it is necessary that the gate openings of the wheel be placed from one to one and one-quarter wheel diameters below the water surface. The head under which the wheel is to operate, however, greatly affects the formation of the vortex. High velocities of flow will facilitate their formation; therefore greater heads will require a greater water covering or other means for the prevention of vortex formation.

As the wheel usually has a greater diameter than the height of the gate it can be set vertically with less danger of air interference than when set horizontally. For this reason the vertical wheels are more readily adapted to low heads and have in the past been more widely used for developments under low and moderate heads.

With both horizontal and vertical wheels the wheel may be protected from the formation of the vortex by a solid wooden float, or may be partially encased or covered with an umbrella-shaped cover the edges of which can be brought below the level of the upper gates of the turbine thus allowing the wheel to be set near the head water surface without the serious interference above mentioned. In all such cases the float or cover must be so arranged as to admit the water to the wheel gates without undue velocity in order to prevent the loss of head. If this is done the efficiency and power of the wheel will not be affected.

Figure 316, page 504, shows the permanent steel umbrella coverings successfully used on the wheels of the Southern Wisconsin Power Company at Kilbourn, Wisconsin, where the depth over the fifty-seven inch wheels at low water is not more than one and five-tenths feet.

238. Arrangements of Vertical Shaft Turbines.—Figs. 317, page 505 and 318, page 506, show twelve typical arrangements of reaction turbines. Diagrams A, B, C and D of Fig. 317 show typical arrangements of vertical wheels. Diagram A is the most common arrangement of the reaction turbine in an open penstock for low head. In this case the wheel is set in a chamber called the wheel pit, the flume,

or sometimes the penstock, and is connected with the head race from which it should be separated by gates. The wheel pits in the smaller plants have commonly been constructed of timber; but in the larger plants, they are usually built of a more substantial character,—often of iron or concrete, usually reinforced. Sometimes two or more wheels are set in a single pit; but in the better class of construction, a pit is supplied for each individual wheel or each unit combination of wheels so that each unit can be cut off from its fellows, disconnected from the

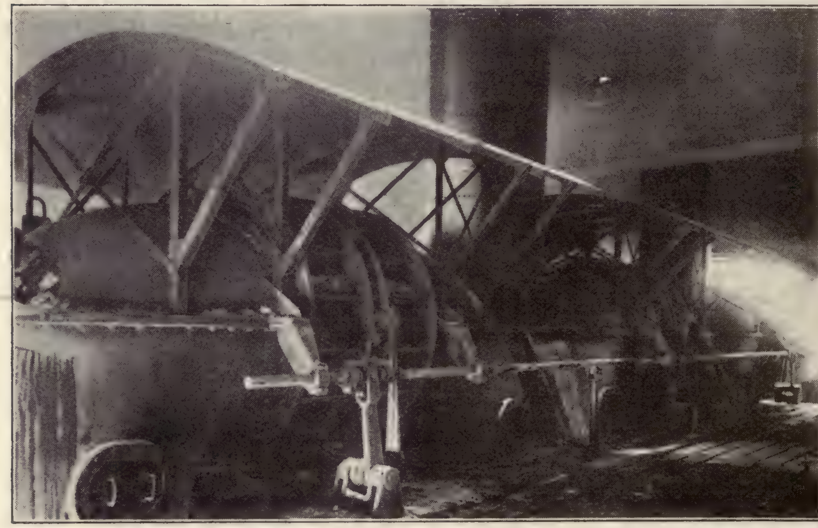

Fig. 316.—Permanent Steel Umbrella Over Wheels at the Kilbourn Plant (see page 503).

transmission mechanism to which it is attached, and examined or repaired without interference with the remainder of the plant. Open pits are commonly used for heads up to eighteen or twenty feet and may be used for considerably higher heads under favorable conditions.

For higher heads, the arrangement shown in Diagram B, or some other form similar thereto, is often found more desirable. In this case closed flumes of steel or reinforced concrete are used, and are connected with the head race by metal, wood, or reinforced concrete pipes to which the term "penstock" is commonly applied. This form of construction permits of the use of vertical wheels with almost any head. In Diagram B the turbine is shown as direct connected to an electrical generator of special design with vertical shaft.

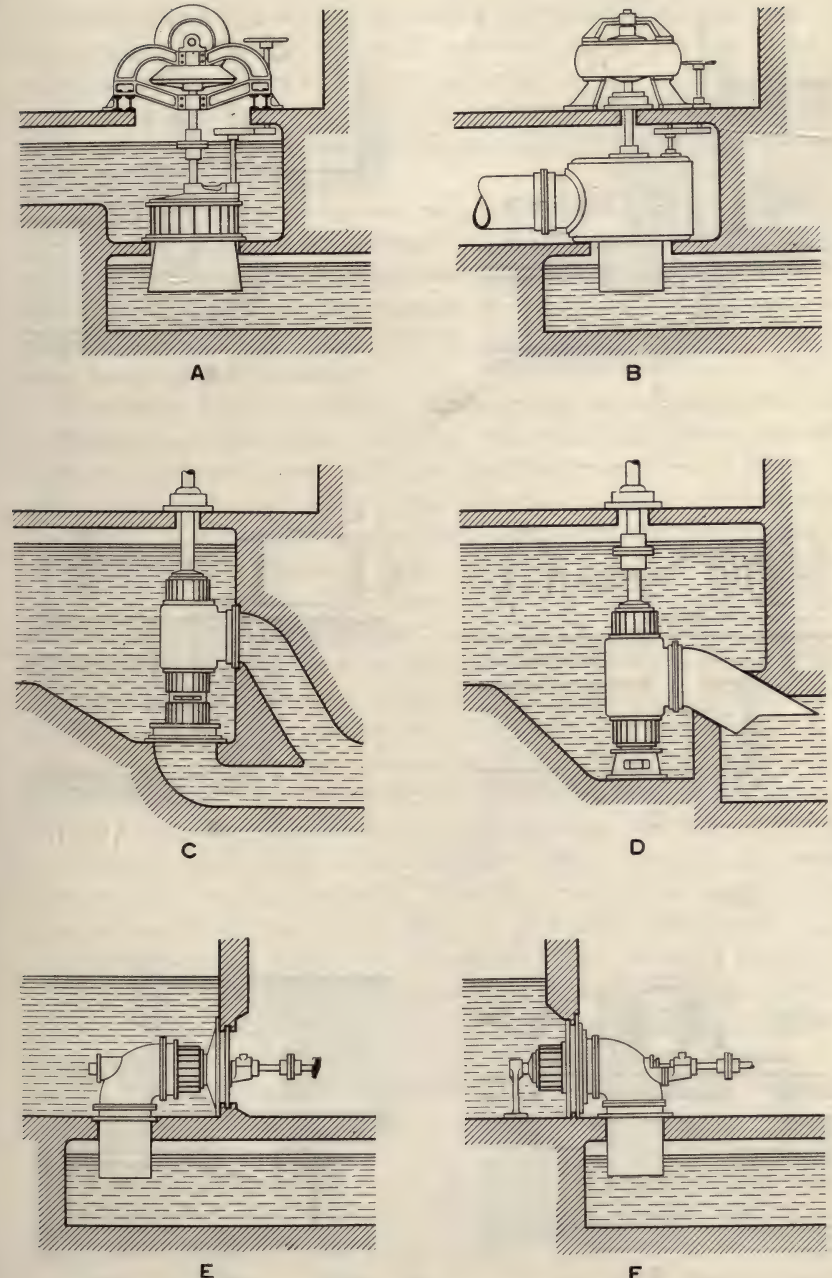

Fig. 317.—Typical Arrangements of Vertical Turbines (see page 503).

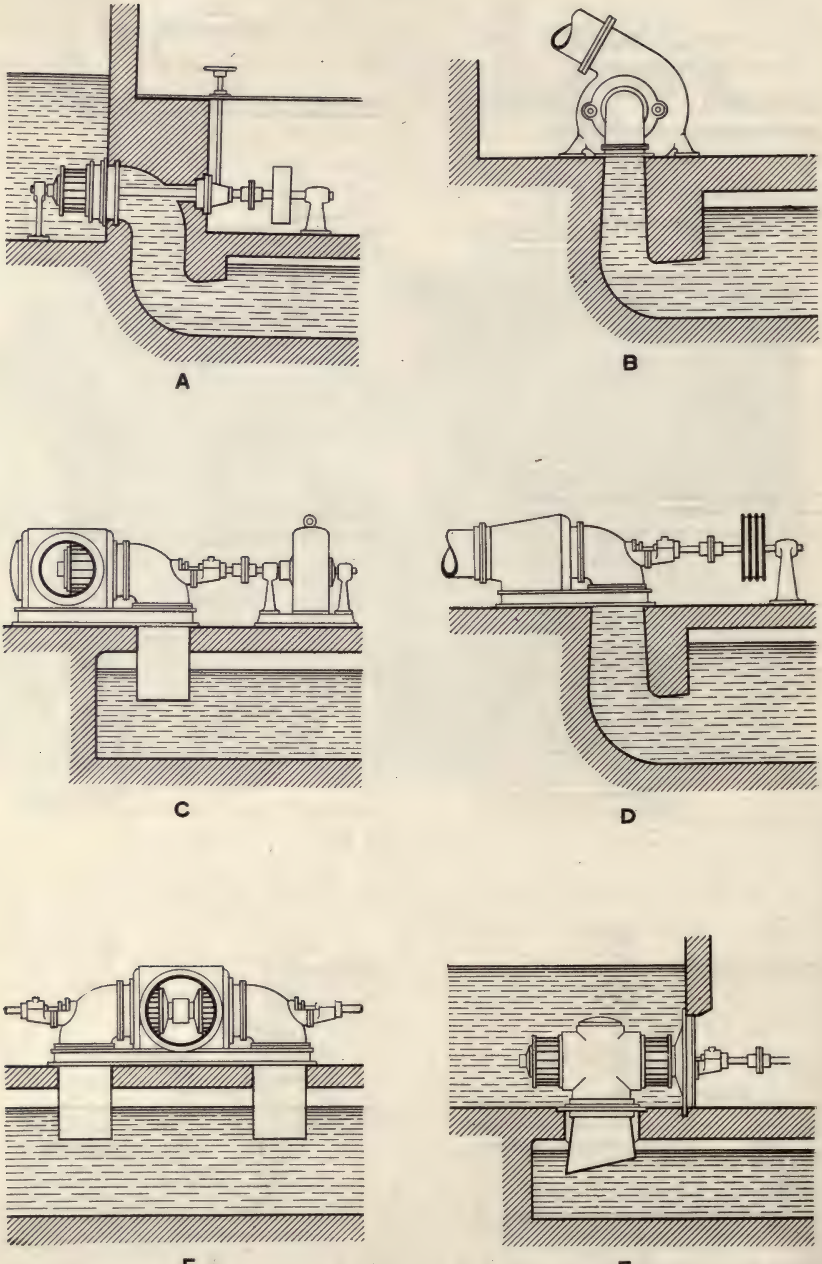

Fig. 318.—Typical Arrangements of Horizontal Turbines (see page 503).

In Diagram A the shaft of the turbine is shown as directly attached to a crown gear which in turn is connected by a spur gear with a horizontal shaft. This horizontal shaft may be direct-connected to a generator as shown in Fig. 342, page 538, or may be attached by belting, ropes, cable or other mechanical means with one or more machines which it is designed to operate.

Diagrams C and D show two vertical types of settings of tandem or multiple wheels. Such arrangements are introduced when it is necessary to reduce the diameter of the wheels on account of increased speed, and at the same time maintain the power of installation by increasing the number of wheels for the purpose of direct connection to some machine to be operated.

In all cases where two wheels discharge into a common draft tube sufficient space is necessary between the wheels to prevent interference and consequent loss in efficiency. The arrangment of wheels in this manner therefore requires a considerable amount of vertical space and, under low or moderate head, involves the construction of a wheel pit of considerable depth in order to secure proper submergence of the upper wheel. This arrangement results in the lower wheel being often considerably below the tail-water and necessitates the use of tail gates and a pumping plant to remove the water in order to make the lower wheels accessible. With this design the plant is made comparatively narrow but the greater depth of construction means an additional expense in the foundation work. Vertical wheels of all types involve a design of satisfactory vertical bearings which are usually less accessible than in the case of horizontal bearings which can be placed at an elevation above the power house floor, and are consequently more readily accessible. The step bearings for single vertical wheels have been long in use and are reasonably satisfactory. The suspension bearing, which is involved in the use of large vertical installations, is not universally satisfactory and, in fact, considerable difficulties have been encountered in so designing a bearing that it will operate without undue expense for maintenance.

239. Arrangements of Horizontal Turbines.—Single horizontal wheels of the common type are shown in Diagrams E and F of Fig. 317 and in Diagrams A, B, C and D of Fig. 318, page 506. In each case the gates of the turbine must be readily accessible to the entering water without undue velocity, and the wheel pit, or penstock, must be designed with this requirement in view.

Diagrams E and F, Fig. 317, and A, Fig. 318, show horizontal types of wheels set in an open wheel pit or penstock.

In Diagram E the wheel has the quarter turn set entirely in the pit, and the main shaft passes through a bulkhead in the wall of the station with a packing gland to prevent the passage of water. In this case the water must flow by the quarter-bend and hence, in order to secure sufficiently slow velocity, the wheel pit must be wider or deeper than in the case shown in Diagram F of Fig. 317. Here the gates of the turbine are placed toward the entering water and the flow is interfered with only by the pedestal bearings which, being placed in the center of the crown or cover plate of the wheel, occupy but little space and offer practically no obstruction to flow.

Diagram A of Fig. 318 is essentially the same in arrangement as Diagram F in Fig. 317, except that in this case instead of a metallic quarter-turn and draft tube, the quarter-turn and draft tube are constructed in the masonry of the power station and the bulkhead is reduced to simply a packing gland through which the shaft enters the power station.

Diagrams B, C and D, Fig. 318, illustrate three methods of enclosing a turbine in a closed flume which is connected with the head water by a closed penstock.

In Diagram B the turbine case is spiral, the water enters tangent to the wheel and at right angles to the shaft and is discharged through a metal quarter-bend into a concrete draft tube.

In Diagram C the water enters the metallic flume in which the wheel is placed at right angles to the shaft, and is discharged through a metal quarter-bend and draft tube.

In Diagram D the water enters the wheel case parallel to the shaft of the wheel and is discharged through a metal quarter-bend into a concrete draft tube.

Diagrams E and F of Fig. 318 show methods of setting horizontal shaft wheels in tandem. Diagram F is for setting in an open flume or penstock. The two wheels discharge into a common shaft chest and use a common draft tube. In Diagram E the wheels have a common closed case or flume connected by a penstock with the head waters and each discharges through an independent quarter-turn and an independent draft tube into the tail-waters beneath. With the closed flume removed, this arrangement can also be used in an open penstock. These diagrams are simply typical of various possible arrangements of wheels that can be adapted with various modifications of detail to meet the local requirements of the engineer for any hydraulic plant which he may be called upon to design.

240. Classification of Wheels.—The classification of the arrangement of wheels as shown in Figs. 317 and 318 may be reviewed briefly as follows:

In this review reference is given to various figures in the preceding and following text in which the type of wheel described is illustrated with more or less modifications.

First: Vertical single wheel, open wheel pit (see Diagram A, Fig. 317, also Figs. 351, page 555, 354, page 558, 355, page 559, and 356, page 561).

Second: Vertical single or tandem wheels in metal casing connected by cylindrical penstock with supply (see Diagram B, Fig. 317, also Figs. 118, page 217, 164, page 255, 321, page 513, and 322, page 514).

Third: Vertical tandem wheels,—two or more wheels in open pit (see Diagrams C and D, Fig. 317, also Figs. 120, page 219, and 357, page 563).

Fourth: Horizontal turbine, open wheel pit, quarter-bend and draft tube within wheel pit,—quarter-bend of metal (see Diagram E, Fig. 317).

Fifth: Horizontal turbine, open wheel pit, quarter-bend, and draft tube exterior to pit,—quarter-bend may be of metal or concrete construction (see Diagram F, Fig. 317, also Diagram A, Fig. 318 and Figs. 327, page 519, and 337, page 526).

Sixth: Horizontal turbine in spiral case at end of penstock, single or double draft tube (see Diagram B, Fig. 318, also Figs. 143, page 232, 146, page 235, and 356, page 561).

Seventh: Horizontal turbine in cylindrical or conical case at end of penstock (see Diagrams C and D, Fig. 318, also Fig. 363, page 571).

Eighth: Tandem horizontal turbines in open wheel pit, single discharge through common or independent draft tubes (see Diagram F, Fig. 318, also Fig. 328, page 520, and Figs. 334 to 341, pages 523 to 529).

Ninth: Tandem horizontal turbine in enclosed cylindrical case with common penstock and common or independent draft tubes (see Diagram E, Fig. 318, also Figs. 13, page 11, 134, page 227, and 330, page 521).

241. Vertical Wheels and Their Connection.—The vertical setting of single wheels is usually the cheapest in first cost, which fact is an important factor that has been largely instrumental in the adoption of this arrangement in most of the older plants. Vertical wheels are most commonly set in open wheel pits. They may, however, be set in a cast iron or steel casing which is then connected to the head

race or dam by a proper penstock. Single vertical wheels can be connected to the machine they are to drive by various means. Belting, transmission ropes, cables, and shaftings, are in common use for such connections. The shaft is usually placed horizontally and is connected by a crown beveled gear and pinion to the wheel. Frequently belts, ropes and cables are connected by pulleys or sheaves to a short horizontal shaft driven in the same manner. When the power of a single vertical wheel is insufficient, two or more may be harnessed by gearing to a line shaft which may be directly connected to the machine or machines to be operated, or otherwise connected as convenience and conditions may require.

242. Some Installations of Vertical Water Wheels.—Figs. 351, page 555, to 354, page 558, inclusive, show the plans, elevations, sections, and details of a small plant of vertical water wheels designed by the writer for the Sterling Gas, Light and Power Company of Sterling, Illinois. The details of this plant are clearly shown by the illustrations and will be discussed at some length later. This plant is located on the Sterling side of the Rock River (see Fig. 375, page 594) and is next to the last plant on the Sterling Race. The head developed is about eight feet and the power of each wheel is about 115 H. P. under this head. Each wheel of the installation is set in an independent pit or penstock which can be closed by means of a flume gate. The wheels are connected to a common shaft extending into the power house and connected with pulleys and belts to the generator.

The plan of the South Bend Electric Company at Buchanan, Michigan, is of similar type and is shown in Fig. 356, page 561. The main shaft is here connected with ten turbines and is in turn directly connected to an electric alternator.

The adaptability of the vertical shaft turbine to low head is well shown in Figs. 319, 320, pages 511, 512. Fig. 319 shows three turbines manufactured by The Trump Manufacturing Company of Springfield, Ohio. These turbines are sixty-one, fifty-six and forty-four inches respectively, and by suitable gearings are connected with a common shaft. These wheels were installed at Bologna, Italy, and operate under a low water head of forty-two inches and under a high water head of twenty-eight inches. It was necessary to set the wheels considerably below the level of the tail-water in order that the turbines should have a sufficient submergence for operation. Fig. 320, page 512, is a similar plant installed at Loches, France. In this case the water is conducted to the turbines by means of a syphon supply pipe in order

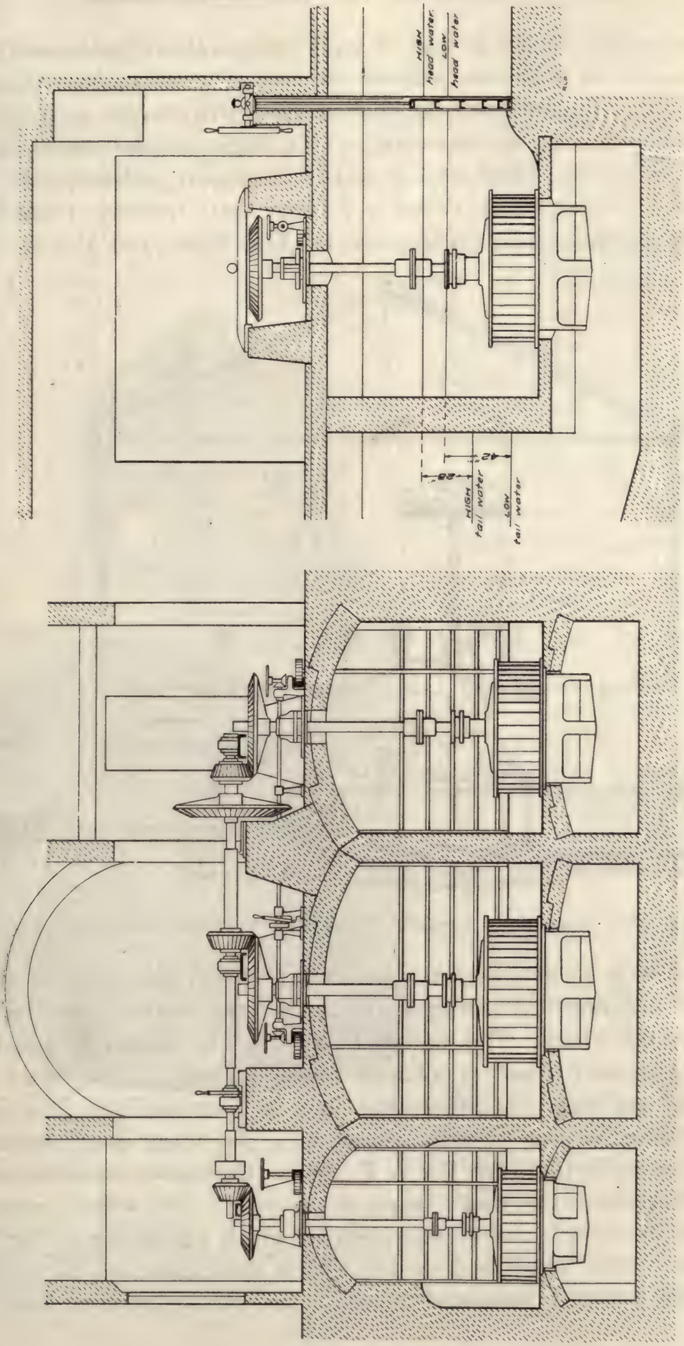

Fig. 319.—Application of Vertical Shaft Turbine Under Low Head (see page 510).

that the turbine might be placed high enough above tail-water that it be accessible at all times without the use of a tail-gate. Air is exhausted from the crown of the syphon by use of a steam ejector whenever the plant is to be started up. This plant operates under the low head of thirty-one inches and is said to work very satisfactorily.

Figure 321, page 513, shows a vertical shaft turbine of the Victor cylindrical gate type manufactured by The Platt Iron Works. This

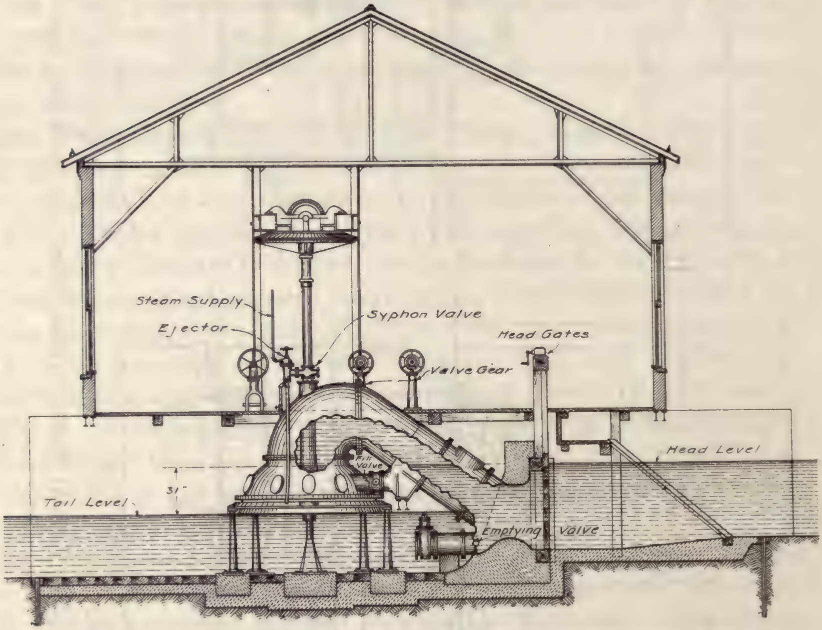

Fig. 320.—Low Head French Water Power Plant (see page 510).

wheel is set in an independent case with provision made for the attachment of a cylindrical penstock conducting the water from the head works to the wheel. This figure shows a special design by which the special generator is set on columns resting directly on the wheel case.

Figure 322, page 514, shows the plant at Trenton Falls, New York, of the Utica Gas and Electric Company. The wheel is a Fourneyron turbine, manufactured by The I. P. Morris Company, operating under a 266 foot head, the water being conducted to the wheel through a penstock the length and arrangement of which are shown in Fig. 385, page 603. The wheel is provided with a draft tube and is regularly connected with the generator above. The moving parts of both ma-

chines are carried by a vertical shaft bearing above the turbine casing and below the generator.

Figure 323, page 515, shows the vertical setting of the 10,000 H. P. turbines of the Mississippi River Power Company at Keokuk, Iowa, with vertical bearing (see Fig. 165, page 256) between the generator and turbine.

Figure 324, page 516, shows the vertical setting of the Pelton-Francis turbines at the Gatun power station of the Panama Canal. In this installation the vertical bearing has been placed above the generator.

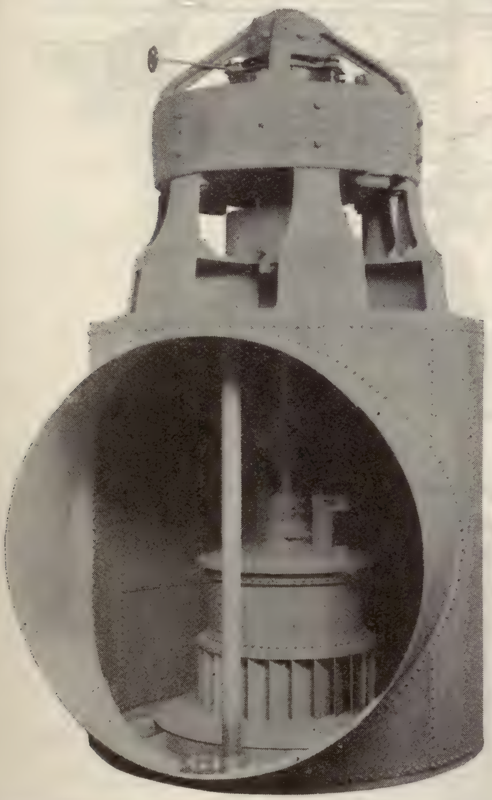

Fig. 321.—Vertical Victor Turbine in Separate Steel Case Made by The Platt Iron Works (see page 512).

243. Some Installations of Vertical Wheels in Series.—In the last three illustrations wheels are shown of sufficient size and operating under sufficient head to be suitable for the independent operation of the machine attached to them. In many cases, however, especially with low head, the arrangement shown in Fig. 319 and in Figs. 342, page 538, to 351, page 555, inclusive, becomes necessary. In such cases considerable loss is entailed by the use of shafts, gearings and belts. These losses are so large that it is desirable to avoid or reduce them if possible. For this purpose vertical wheels are sometimes placed tandem as shown in Diagrams C and D, Fig. 317, page 505. This type of plant is also illustrated by Figs. 325, 326, pages 517, 518, which are illustrative of wheels installed in the plant of the Concord Electric Company, at Concord, New Hampshire.

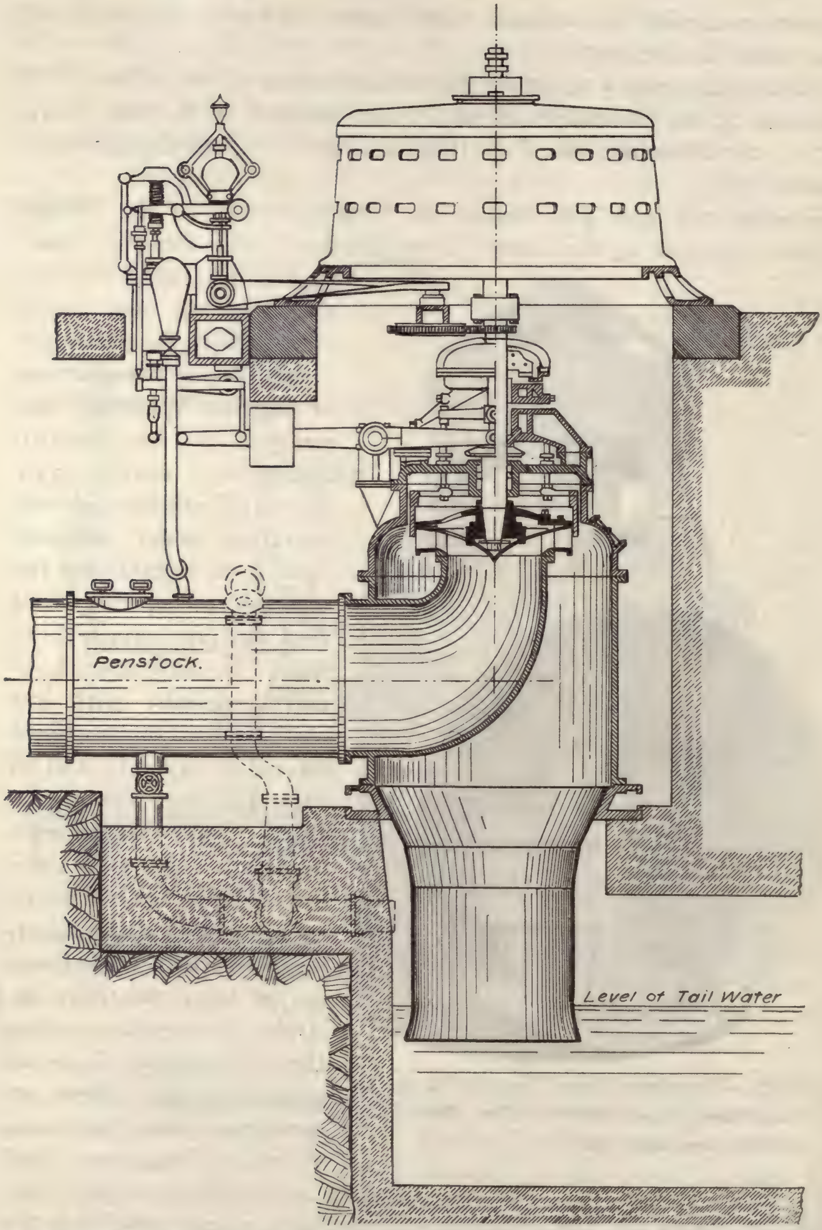

Fig. 322.—The Trenton Falls Plant of the Utica Gas and Electric Co. (I. P. Morris Co.). (See page 512.)

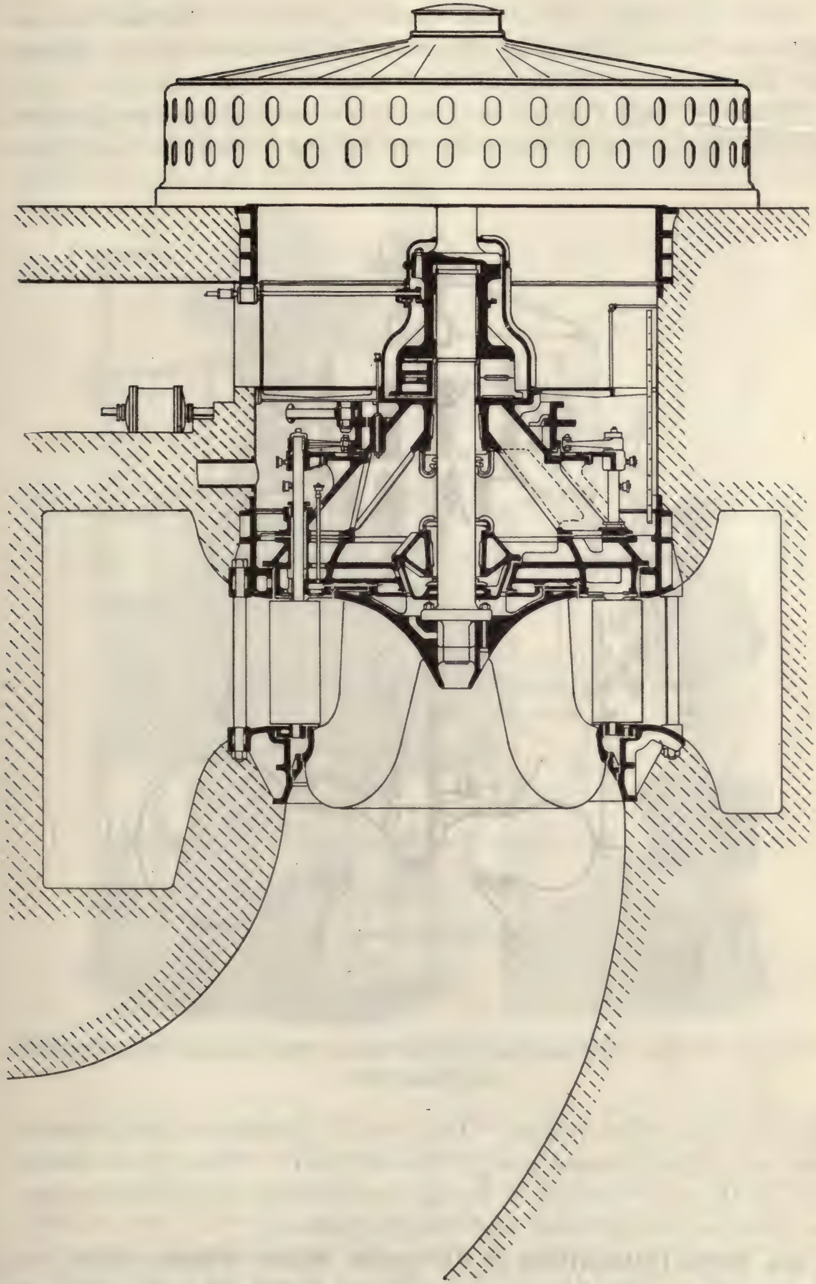

Fig. 323.—Vertical Setting of the 10,000 H. P. Turbine of the Mississippi River Power Co. at Keokuk (see page 513).

Figure 325, page 517, shows tandem wheels for this plant as designed and manufactured by The Allis-Chalmers Company of Milwaukee, Wisconsin, and are described in further detail on page 562.

Figure 326, page 518, is a view of a double vertical unit, designed and built for the Concord Electric Company by The S. Morgan Smith Com-

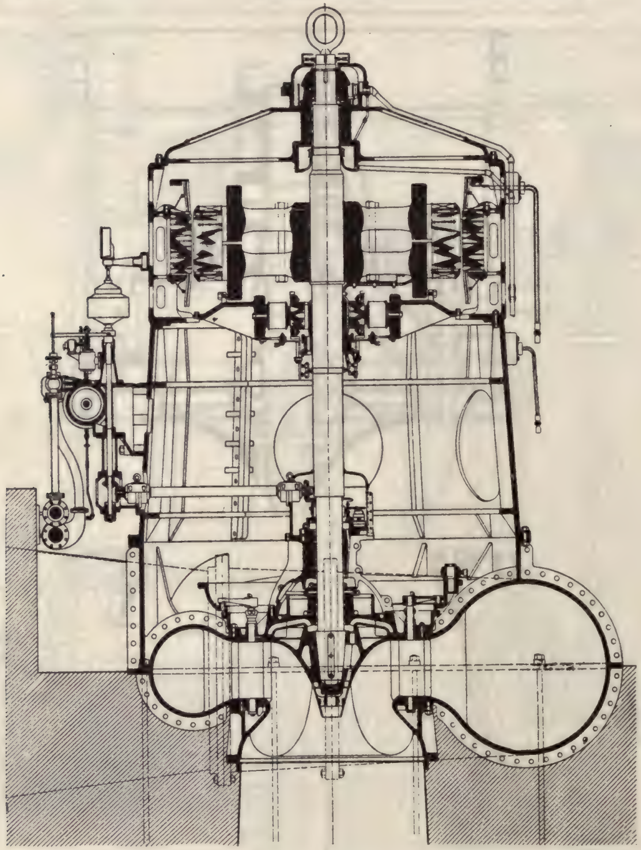

Fig. 324.—Section of Generating Unit at Gatun Hydro-Electric Power Station (see page 513).

pany of York, Pennsylvania. This form of installation has the advantage of a greater concentration of the machinery. This type of installation, while quite common in Europe, is somewhat new in this country and presents several novel and desirable features.

244. Some Installations of Horizontal Water Wheels.—Most machines to be operated by water wheels are built with horizontal shaft,

and, as a direct connection of wheels to the machinery to be operated involves a minimum loss in power and consequent greater efficiency than with the various complicated arrangements often necessary with vertical wheels, the horizontal wheel becomes desirable and is adopted whenever practicable in a modern water power plant. The type of such a plant is well illustrated by the power plant at Turner's Falls,

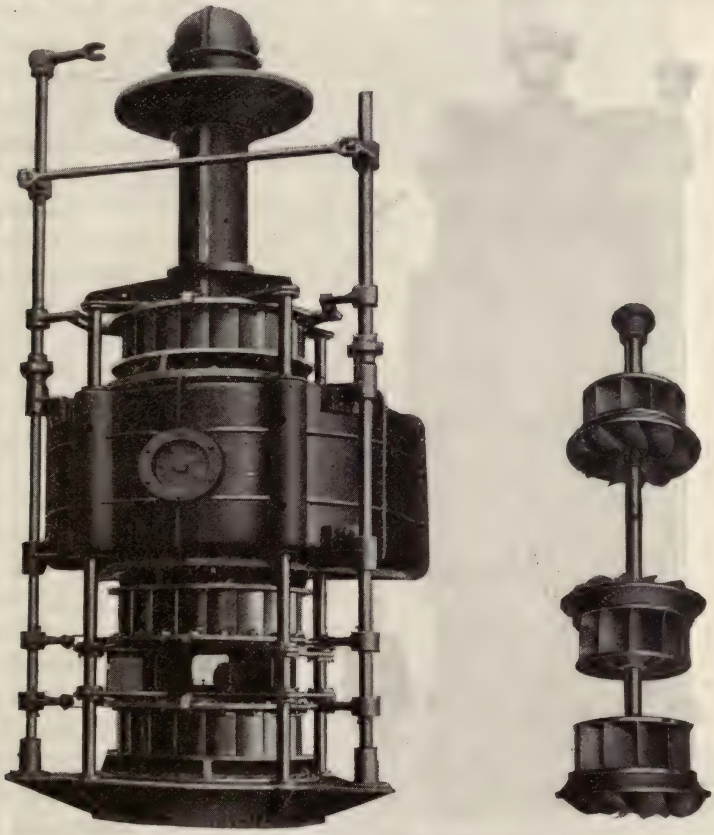

Fig. 325.—Vertical Turbine for Sewall's Falls Plant of the Concord Electric Co. (see page 516).

Massachusetts, shown by Fig. 327, page 519. The single horizontal wheel, direct-connected to the machinery to be operated, is perhaps already sufficiently described in the preceding pages. The arrangement of two or more wheels for such purposes deserves careful consideration. Figs. 328 and 329, page 520, show a plan and section of a double unit, for use in an open penstock, as manufactured by The Dayton Globe Iron Works Company of Dayton, Ohio. These figures

show a plain, cylindrical, draft chest connected with a common draft tube. The details of the arrangement can perhaps be better seen from the half-tone, Fig. 335, page 524, which illustrates two of these units connected together tandem.

Figures 330 and 331, page 521, show a similar double unit manufactured by the same company. This double unit is shown set in a closed flume for connection by a penstock of suitable size with the head works. In Fig. 331 the chest, into which the turbines discharge, is designed so as to give a certain independence to the discharge of the two turbines until they come to the draft chest below the wheel. The turbine case, shown in Fig. 329, seems to have more room than necessary in the upper portion of the case in which interference of the two streams and much eddying are possible, all of which is obviated in the design shown in Fig. 330. The writer knows of no experiments which show conclusively that such loss actually occurred. More information is needed along this line than is now accessible to the engineer.

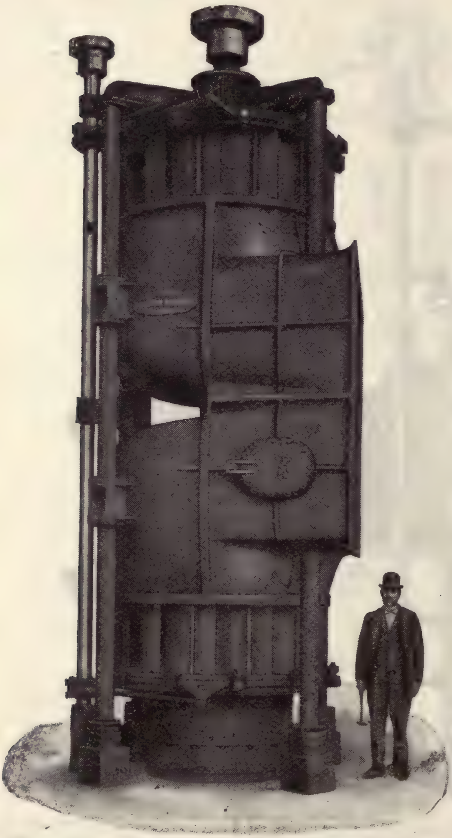

Fig. 326.—Double Vertical Unit Built by the S. Morgan Smith Co.

Figure 332, page 522, shows a shop photograph of a 5,000 H. P. turbine unit built by the Wellman, Seaver, Morgan Company for the Calgary Power Company Ltd. and installed at Horseshoe Falls, Alberta. These units are set in a closed casing, the illustration showing the top and entrance end of the casing removed.

Figure 333, page 523, shows the installation of the Wisconsin Public Service Company at High Falls, Wisconsin, on the Peshtigo River.

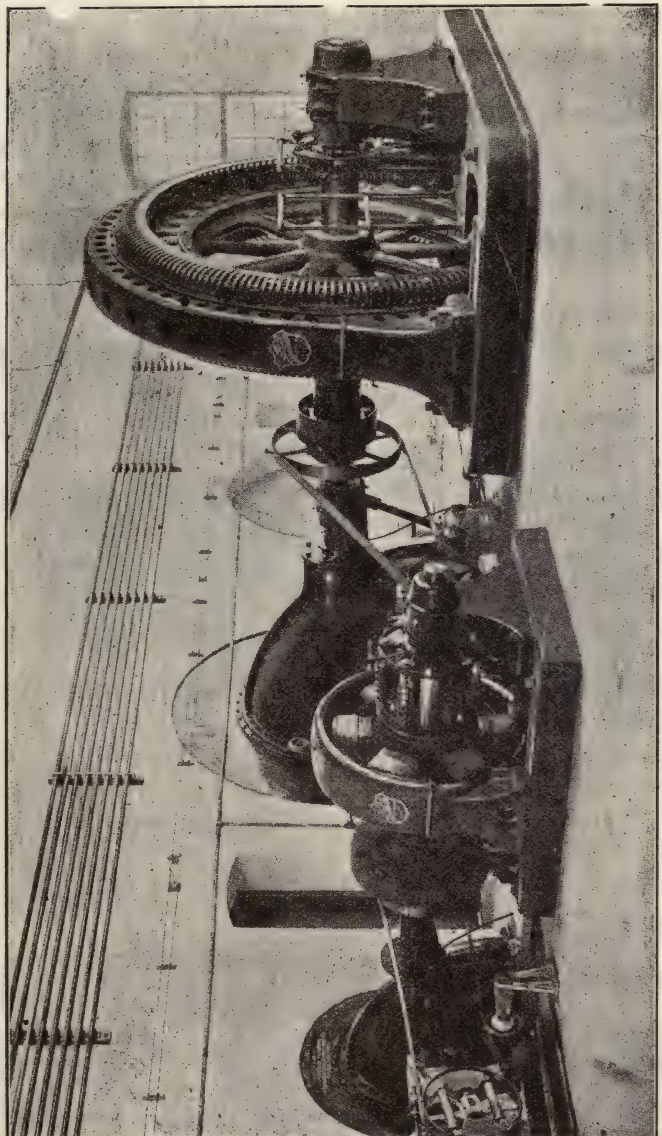

Fig. 327.—Interior of Turner's Falls (Mass.) Power Plant (Allis-Chalmers Co.). (See page 517.)

These units manufactured by the Allis-Chalmers Company are twin turbines of 1,750 H. P. under a head of eighty-two feet. They are provided with fly wheels in order to procure better regulation. Two exciter units of 200 K. W. capacity each are shown in the foreground.

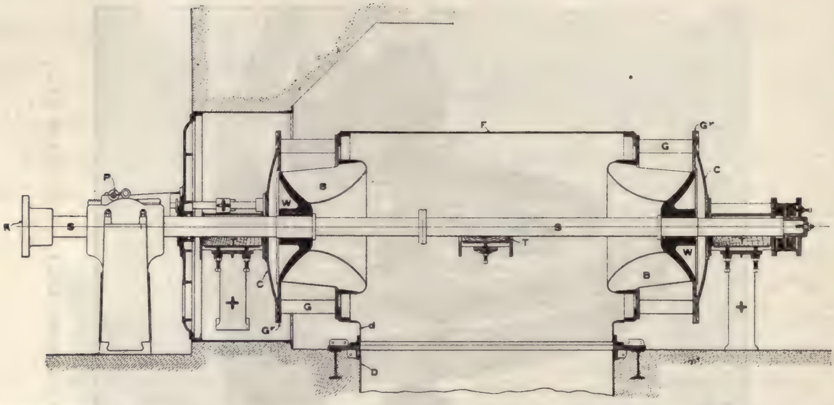

Fig. 328.—Section Double Wheel With Common Draft Tube. Dayton, Globe Iron Works Co. (see page 517).

Figure 334, page 523, is a cross-section of a double unit of the Samson turbine, manufactured by The James Leffel and Company of Springfield, Ohio. This shows a design in which careful attention is given to the maintenance of a uniform and slowly decreasing velocity from the time the water reaches the wheel until it passes from the common draft chest into the draft tube below.

245. Some Installations of Multiple Tandem Horizontal Wheels.— Two double units of the wicket gate type, similar to the double units shown in Figure 328 are illustrated by Figure 335, page 524. These turbines were manufactured by The Dayton Globe Iron Company of Dayton, Ohio, and are shown with the upper portion of the

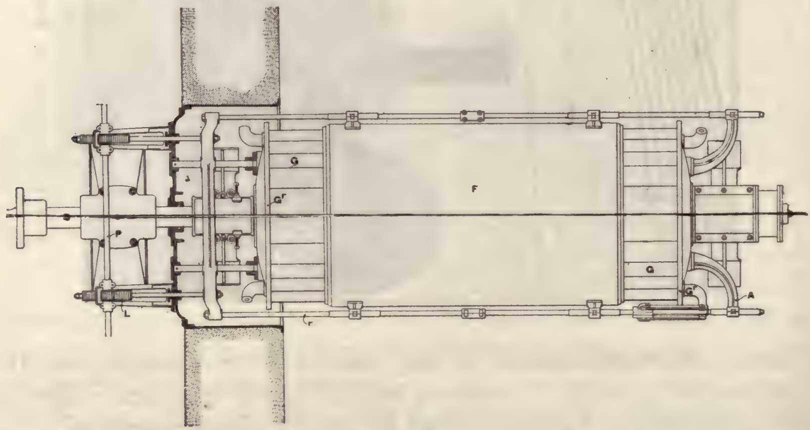

Fig. 329.—Plan of Double Wheel With Common Draft Tube (see page 517).

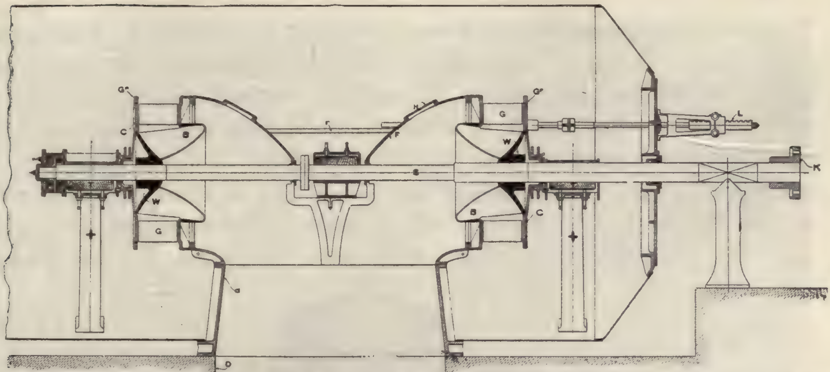

Fig. 330.—Doube Horizontal Turbine in Closed Penstock. Dayton Globe Iron Works Co. (see page 518).

case removed so that the arrangement of the wheels and the gate mechanism are clearly visible. The gates are moved by a cylindrical ring to which each gate is attached independently. The ring is moved by the link connecting the gate ring to the governor rod which, by its rotating, opens or closes the gate as the power needed requires.

Two double units with cylindrical gate, as manufactured by The S. Morgan Smith Company of York, Pennsylvania, are shown in Fig. 336, page 525. The bulkhead casing and the coupling to which the machinery to be operated must be attached, are shown at the left. In this case the governor rods have a horizontal movement, the upper rod moving backward and the lower forward in order to open the cylinder gate.

Figures 337, page 526, and 338, page 527, show a section through one of the main units and a plan of the power house and turbines of

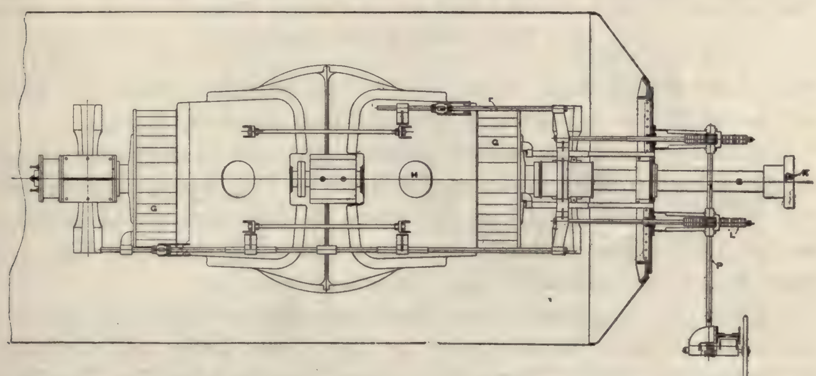

Fig. 331.—Plan of Double Horizontal Turbine in Closed Penstock (see page 518).

The Southern Wisconsin Power Company at Kilbourn, Wisconsin. This plant consists of four main units, each generator having a capacity, at full load, of 1,650 K. W. and an overload capacity of twenty-five per cent. Each unit is direct-connected to six fifty-seven inch turbines constructed by The Wellman, Seaver, Morgan Company of Cleveland, Ohio. Each turbine unit is set in a separate penstock controlled by three independent sets of gates. The four center wheels discharge in pairs into common draft tubes, while the two end wheels

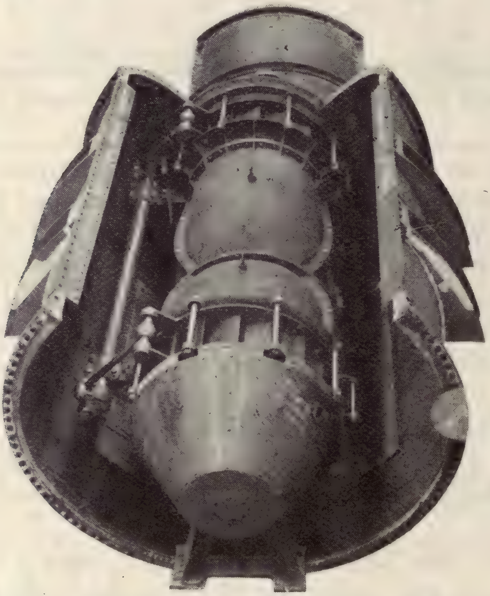

Fig. 332.—5,000 H. P. Turbine Unit of the Calgary Power Co. Ltd. at Horseshoe Falls, Alberta (see page 518).

have independent draft tubes. All of the bearings within the flume are accessible by independent wrought iron manhole casings.

Figures 339 and 340, shown on page 528, are photographs of one of the Kilbourn units taken at the factory. Fig. 339 shows the unit with casing removed and Fig. 340 shows the unit complete as installed, except for the steel umbrella coverings which are shown in Fig. 316, page 504, which was taken after the wheels were set.

Figure 341, page 528, shows four pairs of forty-five inch Samson horizontal turbines manufactured by The James Leffel and Company of Springfield, Ohio. These wheels have been installed for The Penn Iron Mining Company of Vulcan, Michigan, where two such units

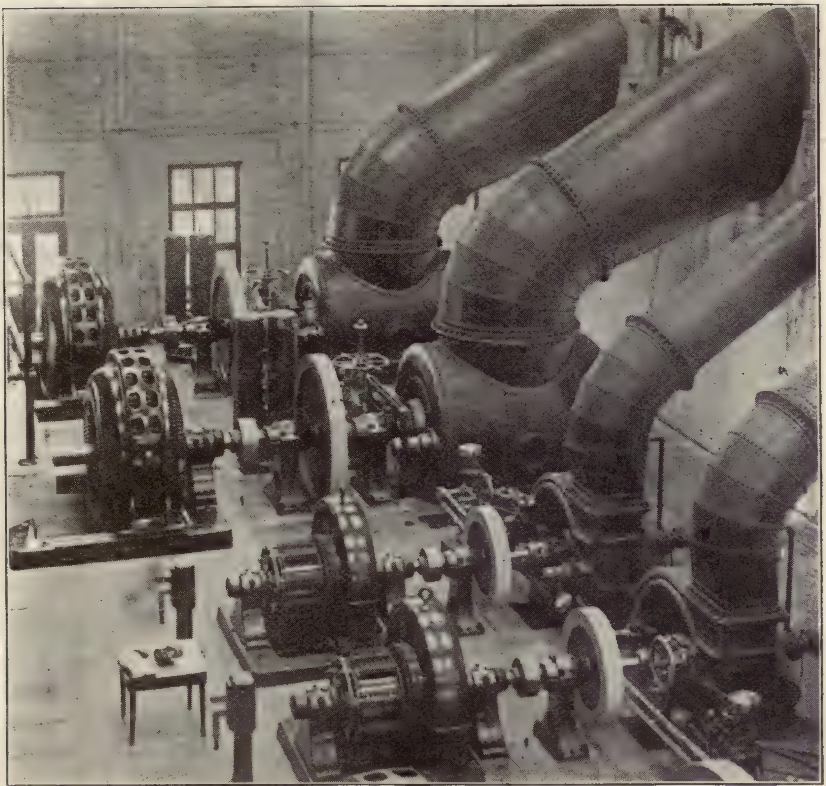

Fig. 333.—Installation of the Wisconsin Public Service Co. at High Falls, Wis. (see page 518).

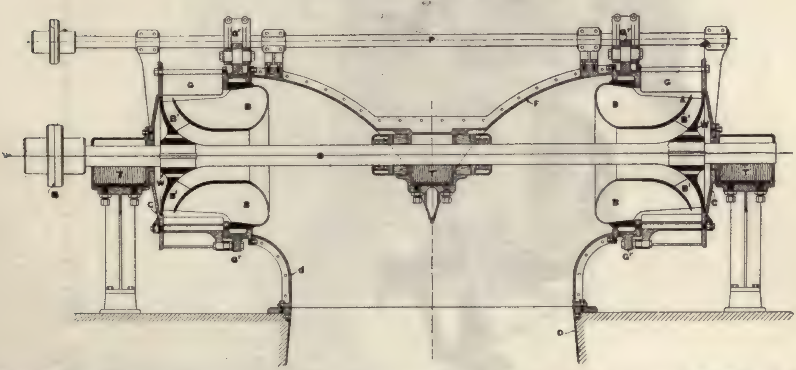

Fig. 234.—Double Horizontal Turbine for Open Penstock. James Leffel & Co. (see page 520).

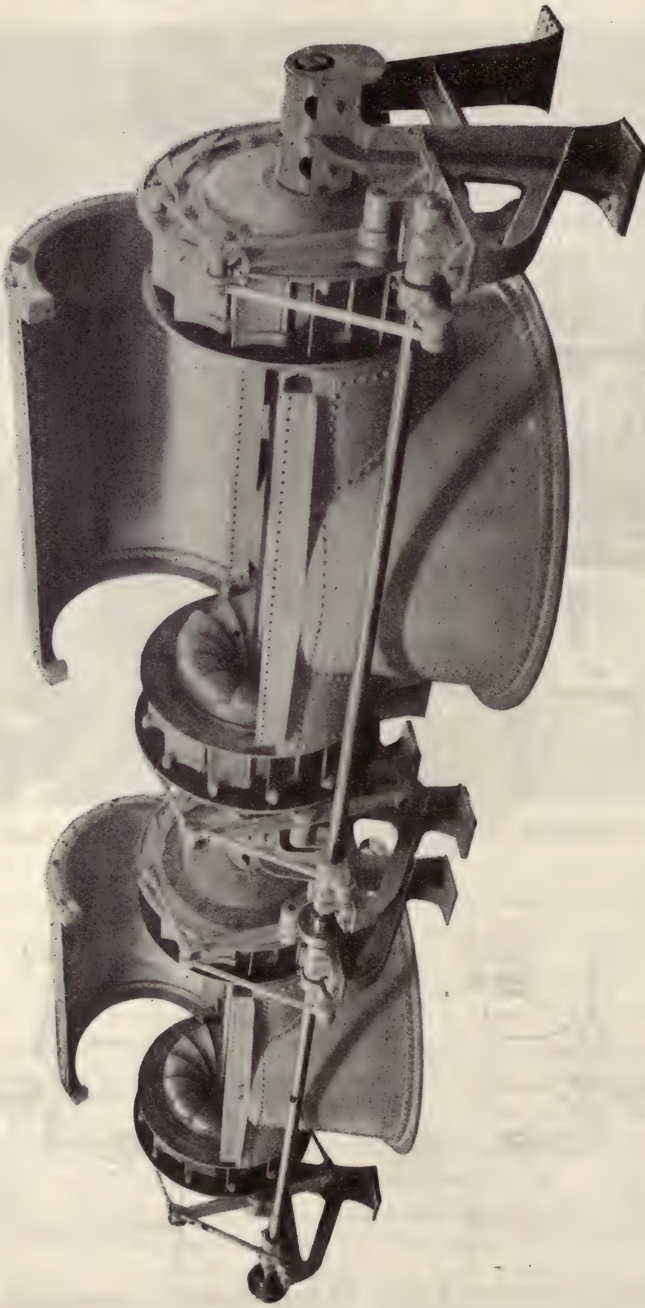

Fig. 335.—Two Double Turbine Units Tandem With Upper Part of Draft Chest Raised (Dayton Globe Iron Works Co.). (See page 520.)

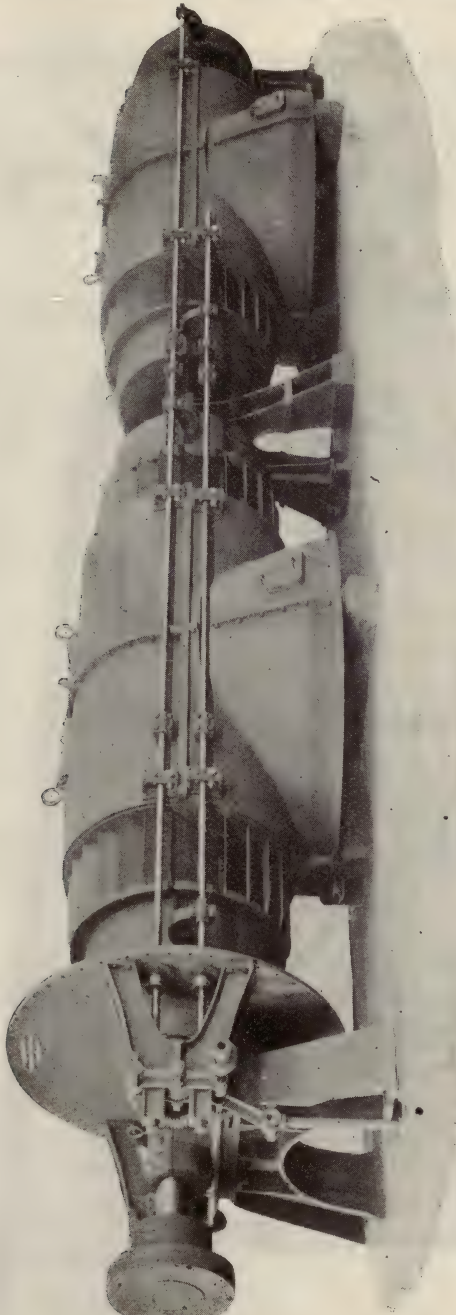

Fig. 336.—Two Double Turbine Units Tandem (S. Morgan Smith Co.). (See page 521.)

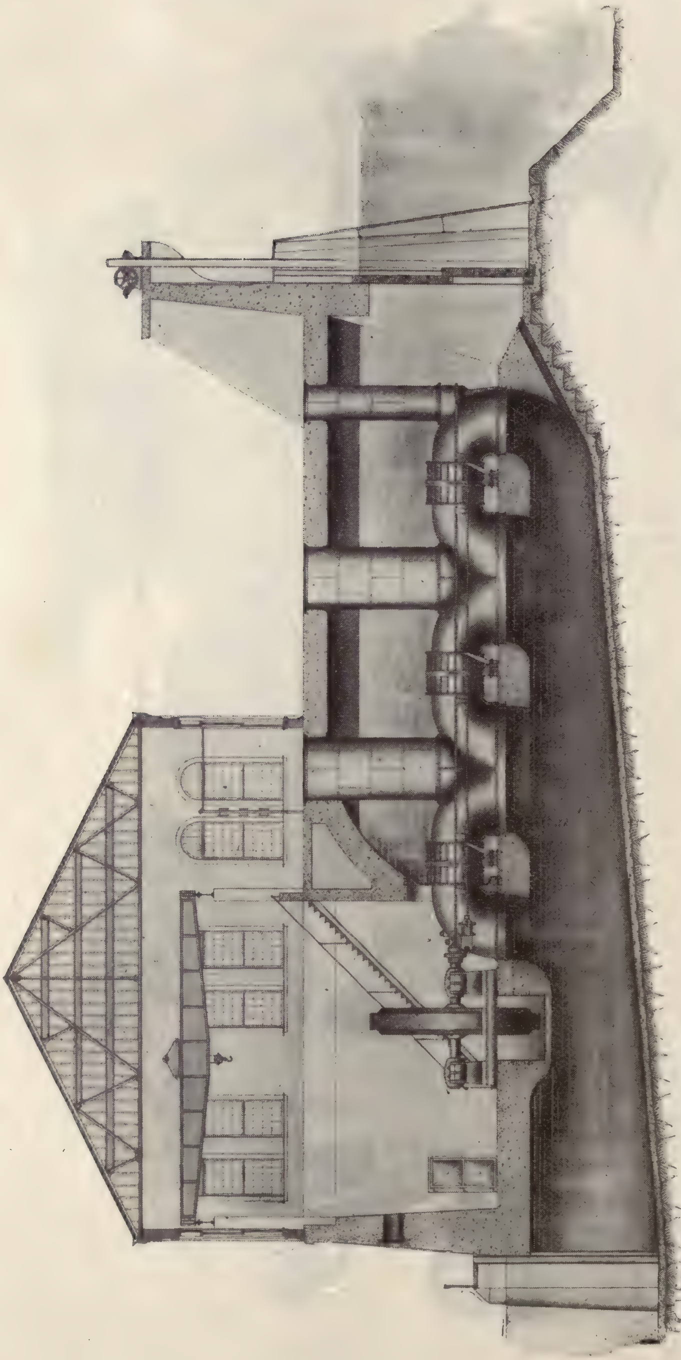

Fig. 337.—Section of Kilbourn Plant, Southern Wisconsin Power Co. (see page 522).

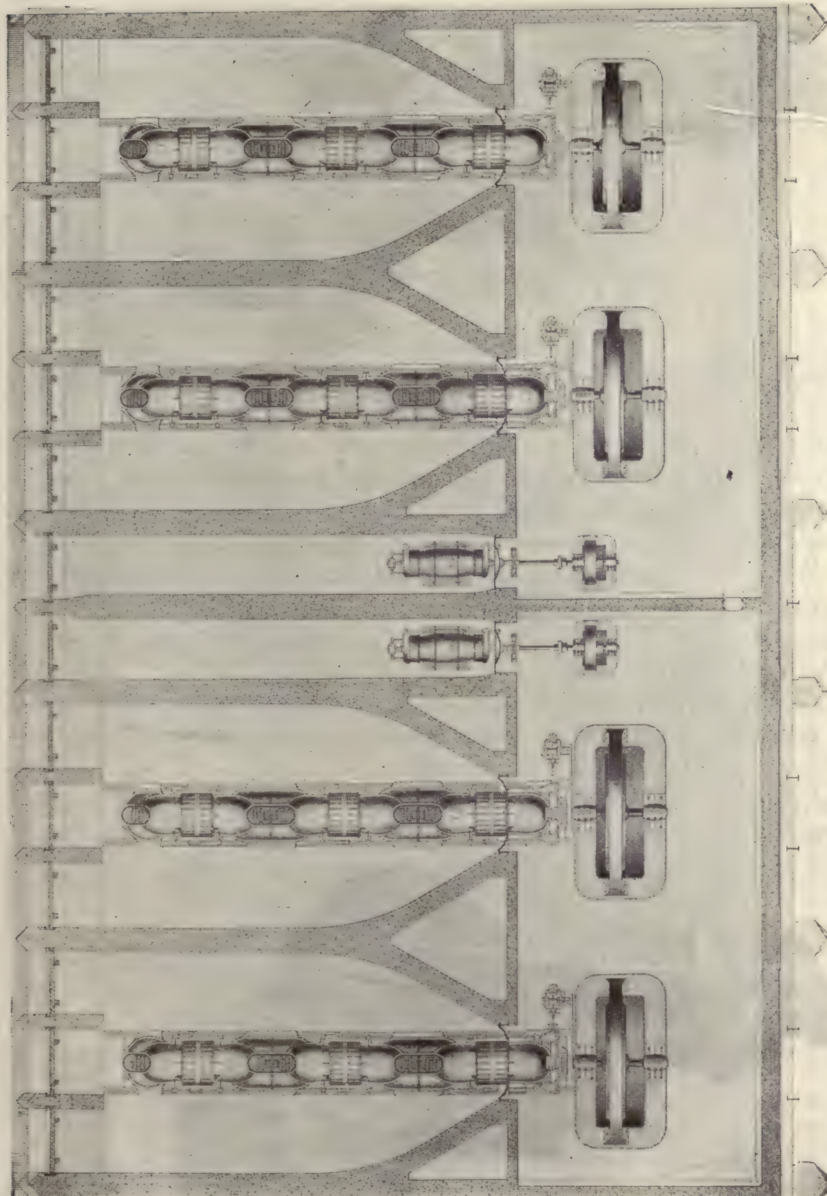

Fig. 338.—Power House of Southern Wisconsin Power Co., Kilbourn. Horizontal Turbines Connected Tandem in Open Penstock (see page 522).

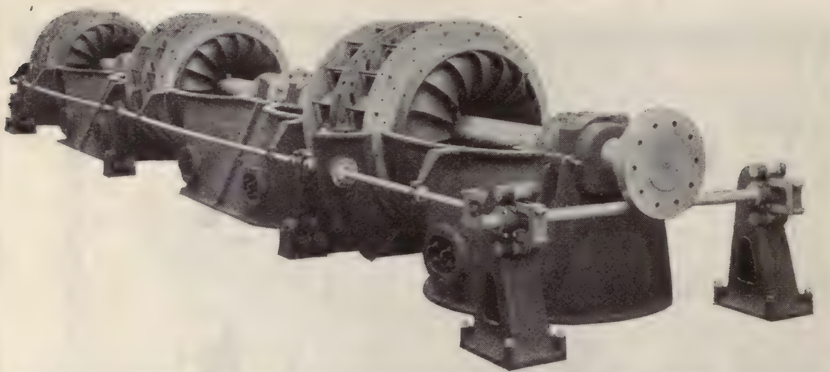

Fig. 339.—Turbine Unit for the Kilbourn Plant Shown With Casing Removed (see page 522).

are now in operation. Eight similar units, designed to deliver 1400 H. P. under fourteen foot head, have been constructed by The James Leffel and Company and are installed in the plant of The Economy Light and Power Company at Dresden Heights, Illinois.

When the head increases above twenty or thirty feet, it may become desirable to convey the water from the head-work by means of a closed penstock as shown in the case of the plant of The Winnipeg Electric Railway Company (see Fig. 362, page 570).

In this plant are shown four wheels in tandem, direct connected to a generator. The bell-mouthed entrance to the penstock should be noticed, also the air inlet pipe which is designed to admit the air into the penstock when the same is to be emptied, and to admit the water gradually and without shock when it is again filled. When the head

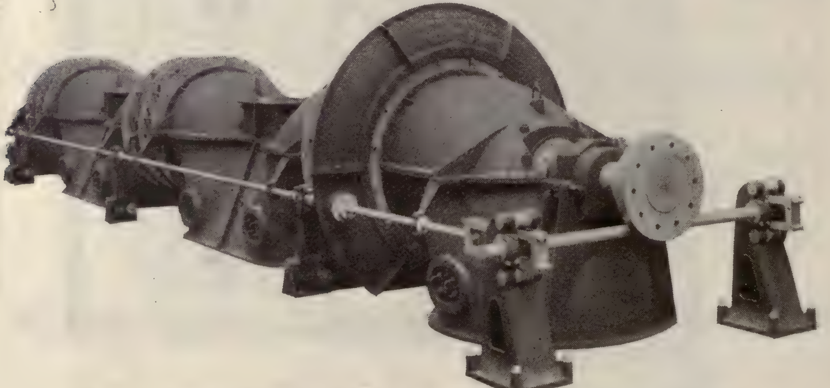

Fig. 340.—Turbine Unit for the Kilbourn Plant (see page 522).

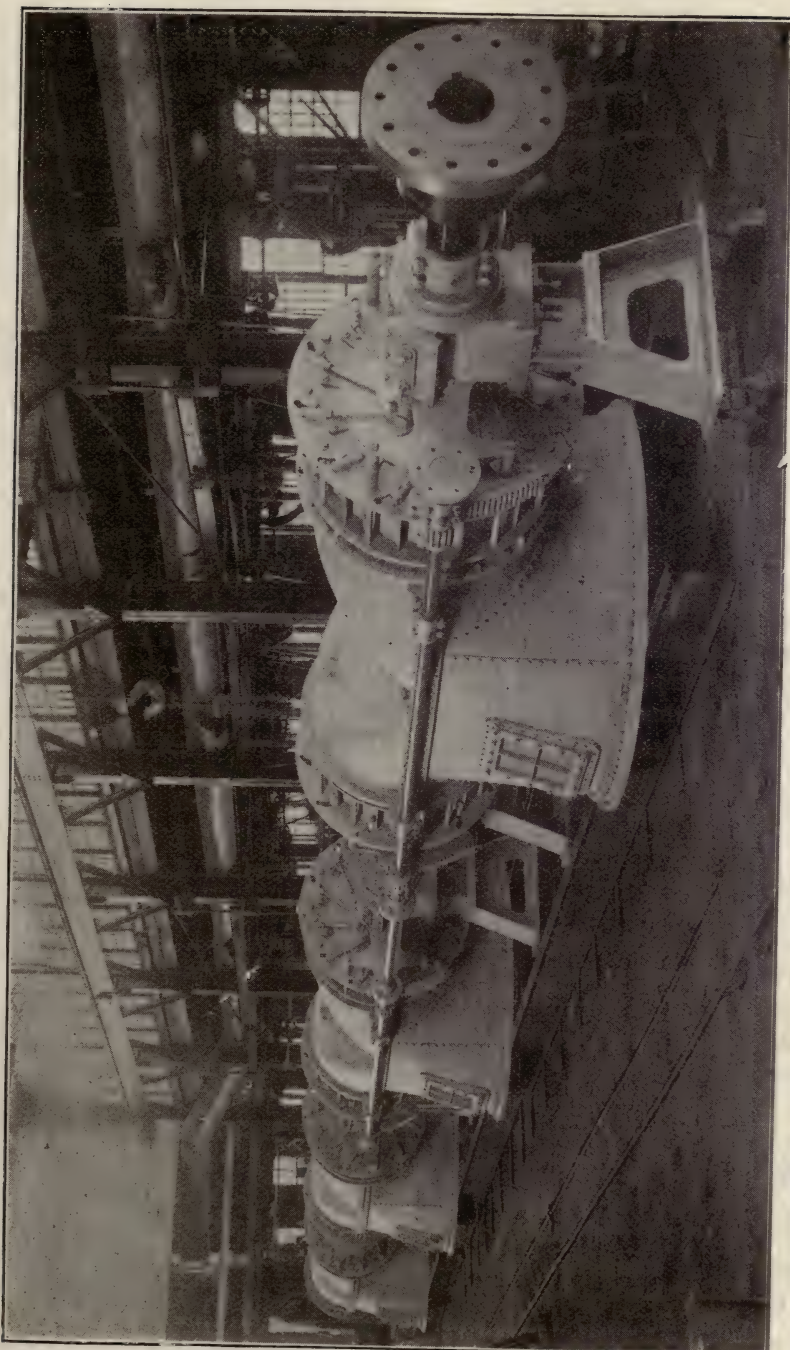

Fig. 341.—Four Pairs 45-inch Samson Horizontal Turbines. 1400 H. P. Under 14-foot Head (James Leffel & Co.). (See page 522.)

becomes still higher the closed penstock becomes imperative as in the case with The Shawinigan Water and Power Company's plant shown in Fig. 367, page 576, where a head of 135 feet is utilized. Similar arrangements and connections for single and double wheels with penstock are those of The Dolgeville Electric Light and Power Company, shown in Fig. 365, page 574, and of The Hudson River Power Company's plant at Spier's Falls, as shown in Fig. 363, page 571.

The plant of the Nevada Power and Mining Company shown in Fig. 371, page 581, involves tangential wheels operating with needle nozzle and discharging freely into the tail-race below.

In the selection and installation of reaction wheels a considerable latitude in the choice and details of arrangement is possible and it is only after a careful examination and consideration of all the conditions of installation that the correct size, speed and arrangement of the wheels can be obtained. Numerous failures, more or less serious, in the past have fully shown the fact that this work demands the most careful attention and investigation of the engineer and should be attempted only after the most thorough study and mature deliberation.

246. Unbalanced Wheels.—In installing horizontal wheels it is usually desirable to use them in pairs with two, four, six or eight turbines in tandem. It is, of course, possible to introduce an odd number of wheels and this is frequently done where it seems to be desirable. There is an advantage in an even number of wheels for in this case the wheels may be, and should be, so arranged as to balance the thrust by the union of a right hand and left hand wheel in each pair. Where an odd number of wheels is introduced, an unbalanced condition arises which can only be taken care of by a thrust-bearing which, at the best, is an additional complication often unsatisfactory and should be avoided if possible.

There is another cause of unbalanced condition which may be here mentioned. If a pair of wheels is so joined together as to use a common draft tube then, on starting the wheel, the vacuum formed in the draft tube is common to both wheels and therefore balanced. If, on the other hand, the wheels have separate draft tubes, when the wheels are started a partial vacuum is commonly created in one of the draft tubes in advance of the other, or even when the wheels are in operation the vacuum in one draft tube is not as great as in the other, creating thereby a thrust in one direction or the other which must be balanced by the connection of the two draft tubes by an air pipe or must be taken up by a thrust-bearing as in the case of a single wheel.

CHAPTER XVII

THE SELECTION OF MACHINERY AND DESIGN OF PLANT

247. Symbols.—The symbols and letters used in the chapter have the following significance:

a = Area.

b = Value of one hydraulic horse power year, delivered at the wheel; cost of penstock and losses in same omitted.

B = Value of power lost per foot of pipe.

c = Cost of pipe per pound in place, in dollars.

C = Cost of pipe per foot in place, in dollars.

d = Diameter of pipe in feet.

f = Friction factor to be estimated from the diameter and rate of flow through the pipe.

g = Acceleration due to gravity.

h = Head of water on the point under investigation.

h_3 = Total friction head.

h′ = Friction head per foot of pipe.

i = Rate of interest, depreciation and sinking fund on the first cost of pipe.

I = Annual interest and depreciation on one foot of pipe.

l = Length of pipe.

L = Total annual loss and expense per foot of pipe = B + I.

P = Horse power.

P′ = Horse power lost per foot of pipe due to friction.

q = Discharge cubic feet per second.

S = Net allowable unit tensile stress of the steel after correcting for efficiency of riveted joints, etc., in pounds per square inch.

t = Thickness of pipe wall in inches.

t′ = Assumed minimum allowable thickness of pipe wall.

v = Velocity of flow in feet per second.

Subscripts used with values of *d*, *t*, etc., denote that these are definite values for the head indicated by the subscript.

248. Plant Capacity.—The selection of machinery for a power plant depends upon numerous conditions. In the first place, for permanent and constant operation, the machinery must be so selected that its total capacity shall be great enough to take care of the maximum load and have at least one unit in reserve so that if it becomes necessary to shut down one unit for examination or repairs, the plant will still be capable of carrying the maximum load for which it was designed.

The desirable reserve capacity of any plant depends on the contingencies of the service or the degree of liability to disabling accident involved in the operation of any plant, and on the relative cost of such reserve capacity and the damages which might be sustained if the plant should at any time become disabled as a whole or in part and incapable of furnishing all or any part of the power for which it was designed. In many manufacturing plants the occasional delays caused by the entire suspension of power on account of high or low water, or for the necessary repair to machinery, are not serious if cheap power is available for the remainder of the year. For the operation of public utilities, and the furnishing of light and power for diverse municipal and manufacturing purposes, the matter becomes more serious and necessitates a sufficient duplication of units to practically assure continuous operation.

For paper mills and other manufacturing purposes water powers are utilized in which the head and consequent power is practically destroyed during high water conditions. For continuous and uninterrupted service such powers are available only with auxiliary power that can be used during such periods. In the same manner reserve capacity may be unnecessary, desirable or absolutely essential as the importance of maintaining uninterrupted power increases.

249. Influence of Choice of Machinery on Total Capacity.—A study of the week day load curve of The Hartford Electric Light Company as shown by Fig. 22, page 46, will show that the load for December, 1901, represents the maximum load which that plant was called upon to carry during the year, and, consequently, was the maximum load for which the machinery must have been selected. A considerable variety of unit sizes would be possible which would fill the requirements of this load curve to a greater or less extent. The maximum or peak load shown in December, 1901, was about 3,000 K. W. If a single machine were selected of 3,000 K. W. capacity for regular operation, then, in order to have one unit in reserve, it would be necessary to purchase two 3,000 K. W. machines or a total capacity of about 6,000 K. W. If, on the other hand, machinery should be purchased with units of 500 K. W. capacity each, it would be necessary to have six of such units in order to carry the maximum load of 3,000 K. W., and a seventh unit of 500 K. W. capacity would be all that would be needed for the reserve. This would give a total capacity to the plant of 3,500 K. W., giving the capacity of the machine purchased some 2,500 K. W. less than the plant first discussed.

250. Effect of Size of Units on Cost.—The cost of machinery is not in direct proportion to its capacity. The larger machinery is somewhat less in price per kilowatt capacity than the smaller machinery. Hence the cost of the last plant suggested would be more than 35/60 of the cost of the first plant. On the other hand, the installation of such a large number of units complicates the plant and is undesirable. For this plant it would therefore be desirable to select five units of 750 K. W. capacity each, or four units of 1,000 K. W. capacity each, giving in one case a total plant capacity of 3,750 K. W. and in the other case of 4,000 K. W.

A plant having units of 750 K. W. or 1,000 K. W. capacity each would have a less total kilowatt capacity and, consequently, a less first cost compared with a plant having units of 3,000 K. W. capacity. Such a plant would also have a less number of units and consequently less complication in the arrangement than a plant having units of 500 K. W. capacity.

251. Overload.—In the above consideration no mention is made of overload capacity. The ordinary direct-current machinery can be operated at about twenty-five per cent. overload for short periods of perhaps one hour at a time without danger to the machinery. Alternating machinery can be operated at fifty per cent. overload at similar times or at twenty-five per cent. overload for two hour periods. In consequence of this condition it is frequently possible to purchase machinery of considerable less capacity than the total load would indicate, depending on the overload capacity of the machine for short periods of maximum load. Unless, however, the estimated load curve covers all possible contingencies for maximum power it is desirable to retain this overload capacity as a provision for a second condition which has not been fully covered in the estimate of the daily load curve; or, in other words, it is desirable to retain the overload capacity as a factor of safety.

252. Economy in Operation.—A second matter that needs the careful consideration of the engineer in the selection of machinery is the question of economic operation under variation in load. A reference to the efficiency curve of most machines will show that the machine will operate most efficiently at some particular load, usually from .75 to full load, and will perhaps give the best results at from .75 to 1.25 load, or to twenty-five per cent. overload. It therefore becomes important to so select machinery that the units will be operated, so far as practicable, at or near their most efficient capacity; and by the start-

ing (or stopping) of additional units, the entire plant can be operated efficiently under all conditions of load.

An examination of the load curve of The Hartford Electric Light Company (Fig. 22, page 46), for the full week day load in March, June, September and December, will show that for securing the most efficient results at all times in the day, and at all times in the season, units of 500 K. W. capacity would apparently be the best. Such units would take care, efficiently, of the minimum loads that occur at 6:00 A. M., between 12:00 and 1:00 P. M., and at about 7:00 P. M. At such times one of these units would operate efficiently; but in most cases the period at which it could be operated singly would be for a few minutes only, or perhaps for an hour at the most, when the additional unit would have to be cut in. A 750 K. W. generator would operate with almost as great an efficiency at these times and it would, with its overload capacity, take care of the load for a much greater period of time each day. The 1,000 K. W. machine would perhaps fulfill these requirements even to a greater degree. While it would be less efficient at the minimum point of the load, it would have the advantage of operating singly for a much wider range of load and the additional advantage that, as a rule, the larger the machine the higher the full load efficiency curve.

The complications resulting from the numerous machines, and the losses entailed thereby, have also to be considered and must be carefully weighed in this connection.

The circumstances of operation and many local conditions, which appertain particularly to the plant in question, must be weighed in connection with the selection of this machinery. There is no definite law by which the selection of machinery for any plant can be reduced to an exact science, and several combinations of machinery are possible in almost any plant and will give reasonable satisfaction.

In the above discussion only units of a uniform capacity have been considered and it is usually desirable, other things being equal, to have similar machines so that a minimum number of repairs and duplicate parts may be kept in stock. On the other hand, if a long, low night load is probable, it may be desirable to install one or more units of a capacity suitable to carry such load efficiently.

253. Possibilities in Prime Movers.—A third matter for the careful consideration of the designing engineer is the possibility of a prime mover that is to be used for operating the machines in question. If a steam or gas engine is to be used as the motive power, there is a wide

range of selection in speed, capacity and economy of such machinery, and, as a general rule, the prime mover may be selected to conform to the generator or other machine that is to be operated thereby. In the selection of water wheels for prime movers the conditions are radically different and the selection of the size and capacity of the units to be operated is often modified or controlled by the water wheels and the conditions under which they will be obliged to operate.

In the selection of the water wheel one of the most important matters is the head and the range of heads under which the wheel will be called upon to operate. While it is possible to select a wheel so that it will operate at almost any reasonable speed under a considerable head, yet the capacity or power of the wheel rapidly decreases in amount with the speed, and if the speed be too high it will be necessary to join two or more wheels in tandem in order to furnish the power necessary to operate the machinery selected. This is perfectly feasible and is done in a great many cases.

254. Capacity of Prime Movers.—It is important to note that if the generator or other machinery is to be operated under overload conditions the maximum power to be generated must be kept fully in mind in the selection of a prime mover. Steam engines can commonly be operated under a wide range of overload conditions. They are usually rated at their most efficient capacity and can sometimes be operated to fifty per cent. above their normal rating, although their economy under such conditions is apt to materially decrease. Gas engines, on the other hand, are commonly rated at very nearly their full capacity and hence the machinery which they are to operate can be operated only to about the normal rated capacity of the engine.

Water wheels are commonly rated at very nearly full gate and consequently at nearly full power. In some cases they are rated at about seven-eighths gate so that a small margin of additional power is available. In the selection of a water wheel, therefore, it is important that a careful study be made of the actual power that the wheel can generate under full gate and at minimum head. This should be sufficient to operate the machinery at its maximum load with at least a small margin of reserve power, for when any prime mover is once overloaded, the speed is reduced and governing becomes impossible until the excess load is removed or additional units are started up in parallel.

255. The Installation of Tandem Water Wheels.—The installation of two wheels set tandem, either horizontally or vertically, and

directly connected with the machine by a common shaft, is very common and this may be increased to four, six, or occasionally to eight turbines. Every additional machine, however, involves the introduction of increased diameter in the shaft, of additional bearings which must be set and held in alignment, and a complication in the design and construction of the machinery which should be avoided wherever possible. The excuse for the attachment of a number of turbines in tandem arrangement, and the complexity of the plant of water wheels installed, lies in the simplification of the machinery to be operated by them, and in the design and arrangement of other portions of the plant. The complications resulting from the installation of a number of wheels in tandem arrangement has led to the general adoption of the simpler (but usually more expensive) single runner installation (see Figs. 323, page 515, and 324, page 516, in many modern high grade large capacity plants. While the expense involved is often warranted by the resulting simplicity in large plants, such expense is often unwarranted in smaller installations. The extent to which the application of any principle is to be carried is a matter of judgment and can be answered only by experience and the consideration of all of the conditions involved in each particular case.

256. Power Connection.—With the turbine, as with every other prime mover, it is important to convey the power to the machine or machinery to be operated as directly as possible. The turbines should be connected as directly as possible to the machinery to be driven without any unnecessary intervention of gearing, shafting, bearings, belts, cables, or other still more complicated methods of power transmission. Every shaft, every gear, every belt, every bearing and every other means of transmission that intervenes between the power generated in the wheel and the machine in which the power is to be utilized means an extra loss and a decrease in the efficiency of the plant. The machine to be operated should, therefore, whenever practicable, be direct connected to the shaft of the turbine instead of being connected with the turbine by any intermediate mechanical means (see Figs. 321, page 513. 327, page 519, and 337, page 526). Direct connection of machinery and turbine involves a careful selection of both machinery and turbine so that both will work satisfactorily at the same number of revolutions per minute. This frequently involves extra expense that may not be justified in plants for many purposes.

Other methods of connection or of power transmission are, therefore, frequently necessary. With many low head installations direct

connections are impracticable for a number of reasons. Sometimes various machines with diverse revolutions are to be driven by the same wheel and the revolutions of the turbines installed must differ from some or all of the machinery to be operated and some form of connection other than the direct must be used. Even where the importance of the plant makes it desirable to use direct connection, it frequently happens that a single turbine gives an insufficient power at the speed desirable for connection to a machine of the desired capacity. Under such conditions it is necessary to unite two or more turbines in order to generate sufficient power for the purposes for which the plant is to be designed. The necessity of using a large number of turbines in a single unit may give rise to very long shafts and a large number of bearings, and the loss due to such an arrangement is sometimes considerable, and if poorly arranged will be almost or quite as inefficient as gearings and shafting well maintained.

257. **Various Methods of Connection in Use.**—The most common form of turbine used is a single vertical turbine, connected by a beveled crown gear and pinion to a horizontal shaft. Several of such turbines are commonly coupled up to the same shaft and may be set in a single or in separate wheel pits. Such types of installation are shown in Figs. 351 to 356, pages 555 to 561. Fig. 342, page 538, shows the turbine harness in the plant of The Oliver Plow Works at South Bend, Indiana, installed by The Dodge Manufacturing Company. The arrangement of the wheel is quite similar to that illustrated by Fig. 356, page 561. Three or four vertical wheels are here each connected by a gear and pinion with a horizontal shaft, which, in turn, is connected to an electric generator. In all such cases more or less energy is lost in transmitting the power through the gearing and numerous bearings to the generator. Sometimes it is found desirable not to connect the generators directly with the main shaft, but to connect the generator or other machines to be operated by the power plant by belting them to driving pulleys attached to the same horizontal shaft, as shown by Fig. 343, page 539, which shows the power plant of The Trade Dollar Mining Company near Silver City, Idaho. This, however, introduces another source of loss through these belts but possesses a certain flexibility due to the ability to thereby drive various small units at a variety of speeds by the simple process of changing the diameter of the pulleys used to drive such machinery. Sometimes rope drives can be used to advantage in place of belts. This is especially true where the distance is great or the alignment

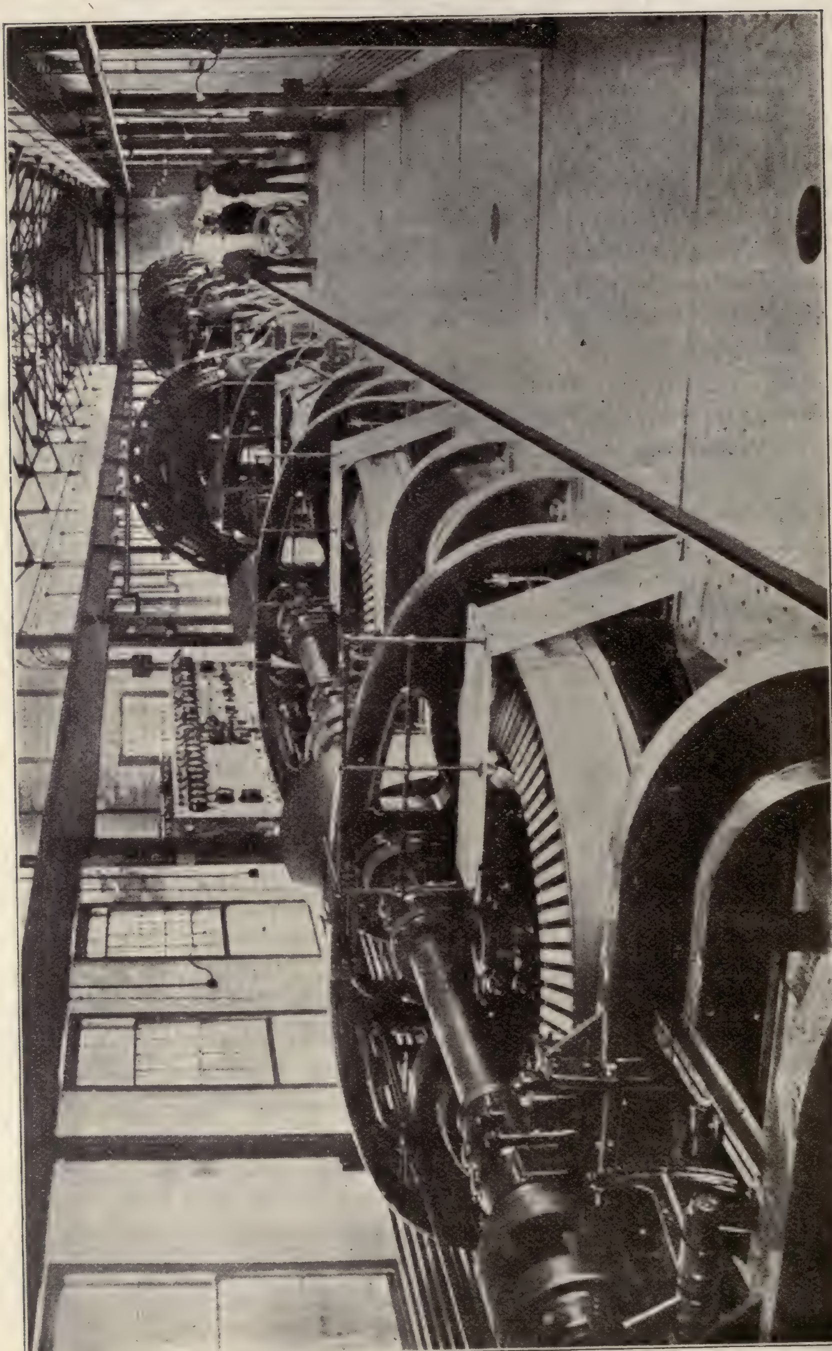

Fig. 342.—Wheel Harness—Oliver Power Plant, South Bend, Ind. (see page 537).

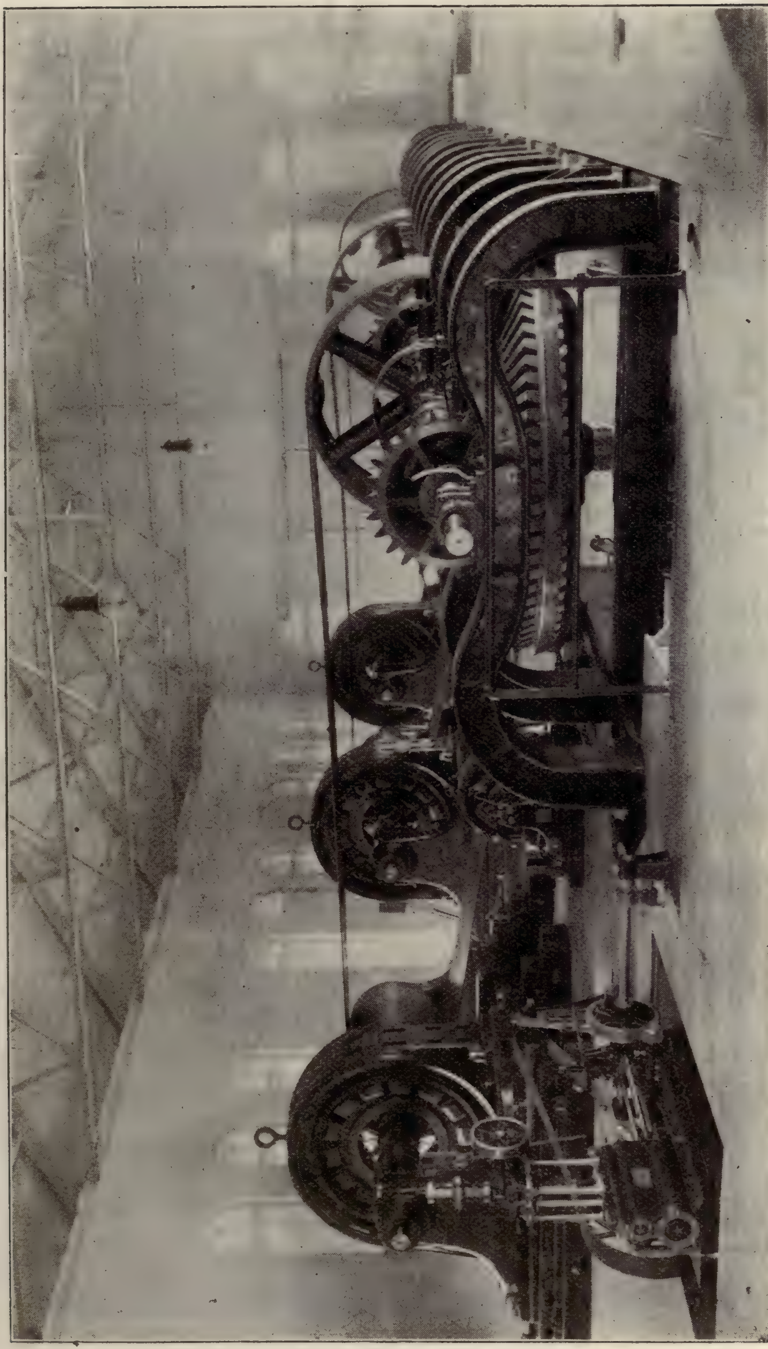

Fig. 343.—Power Plant of the Trade Dollar Mining Co., Silver City, Idaho (see page 537).

other than direct. Examples of such connections are shown by Figs. 344 and 345 on this and the succeeding page.

Direct connected plants are shown in Figs. 321, page 513, 327, page 519, 337, page 526, 342, page 538, etc.

258. Use of Shafting.—A shaft connecting a machine to a prime mover, or imposed in any manner in any power transmission, must be carefully designed and constructed. It must be carefully aligned and

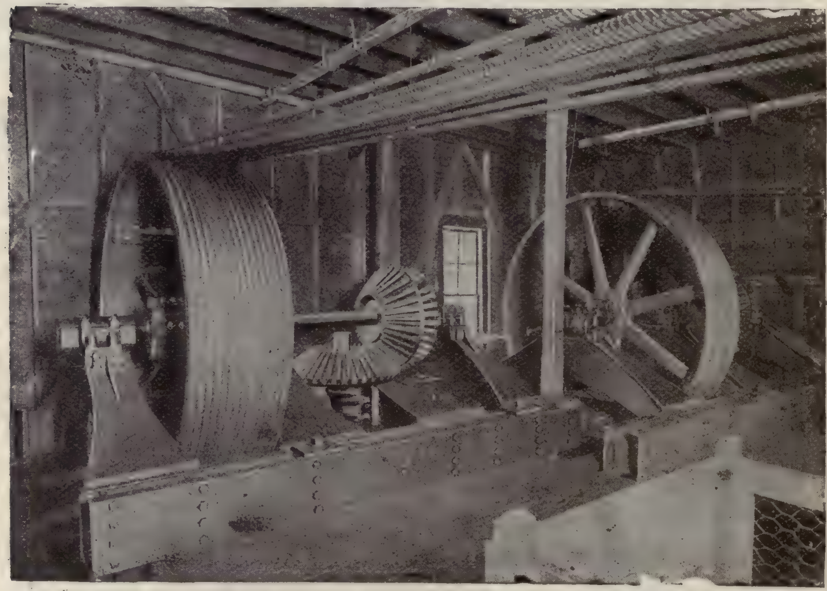

Fig. 344.—Harness and Driving Sheaves, Southwest Missouri Light Co., Joplin, Mo.*

have its bearings carefully adjusted. Each bearing may be considered as a point in the alignment of a shaft, and, as two points determine the direction of a straight line, it will be seen that each additional bearing is objectionable for it increases the difficulty of obtaining and maintaining a satisfactory alignment. When more than two bearings are used each must be brought and maintained in the best practicable alignment, both horizontally and vertically. All bearings must be of sufficient size that the limit of bearing pressure shall not exceed good practice and they must be sufficiently adjustable so that the shaft shall have as complete and uniform bearing as possible over

* Dodge Manufacturing Co., Mishawaka, Ind.

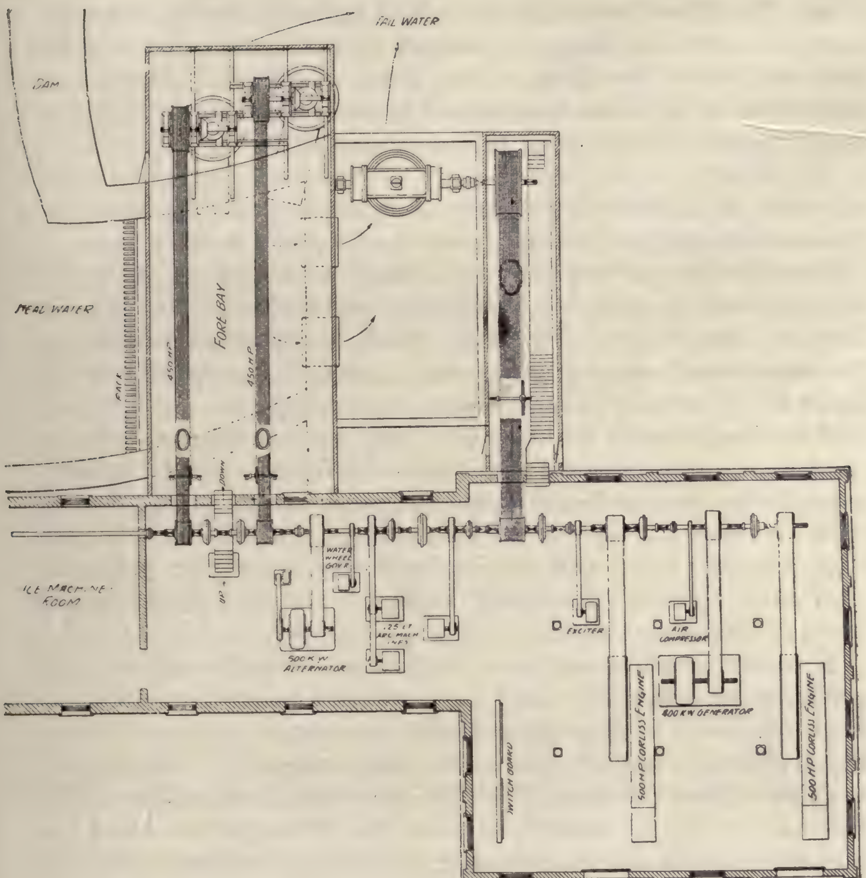

Fig. 345.—Plan Showing Harness, Rope Drive and Jackshaft. Southwest Missouri Light Co.* (see page 540).

the entire surface of the box. Boxes and bearings must be arranged for satisfactory lubrication so that under the hardest service they will not become unduly heated. In order to secure good results the best class of workmanship is necessary and it is also necessary that the plant shall be carefully and properly maintained. A poor shaft, running in poor boxes, poorly aligned, may consume most of the power generated. Shafting, to be reasonably satisfactory, demands frequent and proper inspection, constant lubrication, and proper maintenance or it will soon become a source of great energy loss.

* Dodge Manufacturing Co.

259. The Wheel Setting.—The wheel is usually set in a chamber called the wheel pit, flume, or penstock, which is connected with the head-race either directly or by pipe penstocks or other suitable passages from which it can be separated by suitable gates (see Fig. 352, page 556, and Fig. 356, page 561).

The wheel pit in the smaller plants has commonly been constructed of timber but in most modern plants it is usually built of a more substantial character,—of concrete, plain or reinforced, stone or metal.

Open pits are commonly used for heads up to eighteen or twenty feet, and may be used for considerably higher heads; however, for heads to thirty or forty feet, closed flumes of reinforced concrete are commonly used, and such construction is usually connected with the head-race by metal, wood or reinforced concrete pipes.

When heads exceed thirty or forty feet, and sometimes even under lesser heads, the construction of open wheel pits or closed masonry penstocks becomes undesirable, and the wheels are set in closed metal cases (see Fig. 321, page 513, Fig. 330, page 521, and Fig. 385, page 603) and connected with the headworks by large pipe penstocks. This latter form of construction admits of the use of wheels with heads of almost any height.

A number of wheels can be set in the same wheel pit, and are commonly so set, especially where they are used together to operate a single machine. It is usually desirable, however, to separate the units and set them in separate pits so that any unit can be shut down at any time without interfering with the operation of the plant (see Fig. 338, page 527). The extent to which this arrangement is carried is a matter of policy and depends upon a variety of conditions which the engineer must settle for each particular case.

The arrangement and construction of the wheel pit must be such as to furnish a substantial support for the turbines in order to secure satisfactory operation. In many earlier plants, with wheel pits built of timber, and with the turbine case resting directly on the timber floor, which was often improperly supported, the turbines settled out of alignment and much energy was wasted in undue friction in the transmitting mechanism. The floor or foundation on which the wheel case rests should be of a substantial character and of such a nature that it will not readily deteriorate and allow the wheel to settle. It is usually desirable to support the wheel by a column directly below the wheel case, which should rest upon substantial foundations below the bottom of the tail-race (see Fig. 353, page 557). In all events settle-

ments and vibrations must be prevented or reduced to a minimum in order to eliminate one of the very important causes of loss which is frequently encountered in water power plants. In many cases, due to defects of this kind, water power plants are giving efficiencies at the turbine shaft of fifty per cent. and below, where at least eighty per cent. should be obtained.

260. Trash Racks.—The water entering the wheel pit from the head-race commonly passes through a trash rack the purpose of which is to strain out such floating matter as may be brought by the current down the head-race and which, if not taken out at this point, might float into the wheel gates and if large and heavy enough, might seriously injure the same. These racks usually consist of narrow bars of iron, reaching from above the head waters to the bottom of the wheel pit. These racks must be raked or cleaned out at intervals depending on the amount of leaves, grass, barks, ice or other floating matter in the stream. In water power plants on some streams where large amounts of such floating matter occurs at certain seasons, it is sometimes necessary to keep a large number of men constantly at work keeping the racks clear.

The accumulation of material on the racks will sometimes shut off the entire flow of water if attention is not given to keeping them clear; hence it is sometimes necessary to so design the racks and their supports that they may sustain the entire head of water.

The racks are usually made of rectangular bar iron one-fourth inch by three inches in dimensions, spaced one and one-half inches to two inches, held apart by spools between each pair of bars and held together by bolts passing through the spools and joining together such a number of bars as may be convenient for handling. The spools should usually be placed near the back of the bar so as to allow the rake teeth to pass readily. A tapering instead of a rectangular cross-section for rack bars has been used with a consequent improvement of the hydraulic conditions. The rack should be placed at an angle, as vertical or nearly vertical racks are difficult to rake. The deeper the water, the greater should be the inclination, as with long racks, and especially with high velocities, the clearing of the racks becomes more difficult.

Chain racks and automatic mechanical racks have been attempted but usually without satisfactory results.

Where trouble occurs from ice, involving much winter work, it is frequently desirable to cover the racks with a house in order to protect the workmen.

261. Friction Losses in Penstock Pipes.—From experimental investigations the loss of head due to friction in a pipe under pressure, has been found to vary approximately in accordance with the following principles:

(a) Increases nearly with the square of the velocity; that is, it varies with the velocity head $\frac{v^2}{2g}$.

(b) Increases with the length of pipe.

(c) Is inversely proportional to the diameter.

(d) Is independent of the pressure.

(e) Increases with the roughness of the surface.

These principles may be expressed by the equation

$$h_3 = f \frac{l}{d} \frac{v^2}{2g} \qquad (216)$$

in which f is a factor to account for the roughness and other unknown conditions.

The values of f from experimental investigations on various kinds of pipe are given in Table 38, page 545.

The references given in the table are abbreviated in accordance with the following:

Proceedings of the Institution of Civil Engineers............Proc. Inst. C. E.
Transactions of the American Society of Civil Engineers.........A. S. C. E.
Journal of the Association of Engineering Societies................A. E. S.
Darcy; *Recherches Experimentales Relatives au Mouvement de l' Eau dans les Tuyaux 1857*......................................R. E.
Herschel, Clemens; *115 Experiments on the Capacity of Pipes (1897)*.Herschel

262. Maximum Power From Penstock Pipes.—The theoretical power that can be generated from the water delivered through any pipe line will be the quantity of water q, multiplied by the net head, which will equal the total head h, less the friction head h_3, and divided by eight and eight-tenths (see Section 16).

$$P = \frac{q\,(h - h_3)}{8.8} \qquad (217)$$

Introducing the value of h_3 from equation (216) there results

$$P = \frac{vah}{8.8} - \frac{v^3 fla}{8.8d2g} \qquad (218)$$

The maximum theoretical power that can be delivered by this pipe line can be determined by differentiating equation (218) relative to v and placing the first differential equal to 0, i. e.

$$\frac{dP}{dv} = \frac{ah}{8.8} - \frac{3v^2 fla}{8.8d2g} = 0 \qquad (219)$$

TABLE 38.

*Experimental Values of Friction Factors for Pipes and Conduits.**

Kind of Pipe.	Condition or Coating.	Diameter, inches	Age, Years.	Velocity in Feet per Second.														Reference.
				1.0	1.5±	2.0	2.5±	3.0	3.5±	4.0	4.5±	5.0	5.5±	6.0	6.5±	8.0	10.0	
Cast Iron....	Coal Tar	12	New	.0247	.....	.0212	.....	.0195	.....	.0184	.....	.0173	.....	.....	.....	.....	.....	Williams, Hubbell, Fenkell—A. S. C. E.—47.
	....	12	"	.0303	.....	.0285	.....	.0243	.....	.....	.....	.....	.....	.....	.....	.....	.....	Hamilton Smith—Hydraulics.
	....	12	Old	.0267	.....	.0340	.0365	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	Thos. Duncan—Proc., Inst. C. E.—1853.
	....	16	New	.0181	.....	.0162	.....	.0155	.....	.0148	.....	.0144	.....	.....	.....	.....	.....	Williams, Hubbell, Fenkell—A. S. C. E.—47.
	Varnish	16	"	.0217	.....	.0205	.....	.0191	.....	.....	.....	.....	.....	.....	.....	.....	.....	Hamilton Smith—Hydraulics.
	Uncoated	20	"	.0229	.....	.0213	.....	.0198	.....	.0191	.....	.....	.....	.....	.....	.....	.....	Darcy—R. E. Series 22.
	Coal Tar	30	"	.0181	.....	.0173	.....	.0167	.....	.....	.....	.....	.....	.....	.....	.....	.....	Williams, Hubbell, Fenkell—A. S. C. E.—47.
	Pitch	48	1/8	.....	.....	.0140	.....	.0139	.0137	.....	.....	.....	.....	.....	.....	.....	.....	A. F. Bruce—Proc., Inst. C. E.—123.
	"	48	2	.0153	.....	.....	.....	.....	.0134	.0129	.....	.0127	.....	.0124	.....	.....	.....	Stearns—A. S. C. E.—14.
	"	48	17	.0225	.....	.0229	.....	.0232	.0232	.....	.....	.....	.....	.....	.....	.....	.....	Fitzgerald—A. S. C. E.—35.
	"	48	17	.0213	.....	.0213	.....	.0217	.0217	.0217	.....	.0217	.....	.0217	.....	.....	.....	" " "
	"	48	17	.0148	.....	.0140	.....	.0135	.....	.....	.....	.0217	.....	.0124	.....	.....	.....	" " "
	Coated	61	New	.....	.....	.....	.....	.....	.....	.....	.....	.0113	.....	.0110	.0108	.....	.....	" " "
Riveted Steel	Galvanized	3	"	.0403	.....	.0375	.....	.0364	.....	.....	.....	.....	.....	.....	.....	.....	.....	Williams, Geltner, Ketchum—A.E.S.—Vol.26.
	Asphalt	4	"	.0291	.....	.0263	.....	.0247	.....	.0237	.....	.0229	.....	.0225	.....	.0216	.0209	Schoder, Gehring—Eng. Rec.—1908.
	"	6	"	.0257	.....	.0229	.....	.0217	.....	.0209	.....	.0201	.....	.0194	.....	.0187	.0181	" " " " "
	"	6	"	.0304	.....	.0291	.....	.0279	.....	.0273	.....	.0265	.....	.0262	.....	.0257	.0251	" " " " "
	Black	11	5	.....	.....	.....	.....	.....	.....	.....	.0220	.0220	.....	.0213	.....	.0201	.0194	Hamilton Smith—Hydraulics.
	Asphalt	11	New	.....	.....	.0213	.....	.0194	.....	.0184	.....	.0175	.....	.0167	.....	.0157	.0150	Darcy—R. E.—Series 10.
	"	30	"	.0285	.....	.0242	.....	.0225	.....	.0204	.....	.0195	.0187	.....	.....	.....	.....	Herschel.
	"	42	"	.0252	.....	.0233	.....	.0225	.....	.0220	.....	.0217	.....	.....	.....	.....	.....	"
	"	48	"	.....	.....	.....	.....	.0252	.....	.0237	.....	.0233	.....	.....	.....	.....	.....	"
	"	48	"	.....	.....	.0217	.....	.0209	.....	.0205	.....	.0201	.0197	.....	.....	.....	.....	"
	"	48	4	.....	.....	.....	.....	.0257	.....	.0243	.....	.0233	.....	.0225	.....	.....	.....	"
	"	48	4	.....	.....	.....	.....	.0297	.....	.0279	.0273	.....	.....	.....	.....	.....	.....	"
	"	72½	New	.0220	.....	.0220	.....	.0220	.....	.0220	.....	.....	.....	.....	.....	.....	.....	Marx, Wing, Hoskins—A. S. C. E.—Vol. 40.
	"	72½	2	.0365	.....	.0303	.....	.0267	.....	.0247	.....	.0233	.0228	.....	.....	.....	.....	" " " "
	Rusty	103	5	.0187	.....	.0209	.....	.0220	.....	.0229	.0233	.....	.....	.....	.....	.....	.....	Herschel.
Redwood	Clean	14	New	.0225	.0201	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	Adams—A. S. C. E.—1898—Vol. 40.
Fir.	"	44½	"	.....	.....	.....	.....	.....	.0209	.0201	.....	.0191	.....	.....	.....	.....	.....	Noble—A. S. C. E.—Vol. 40.
	Some Growth	56⅛	"	.....	.....	.....	.0187	.0175	.....	.0162	.0155	.....	.....	.....	.....	.....	.....	" " " "
Fir.	Clean	72½	"	.....	.0229	.0197	.....	.0175	.0167	.....	.....	.....	.....	.....	.....	.....	.....	Marx, Wing, Hoskins—A. S. C. E.—Vol. 40.
Fir..........	"	72½	2	.....	.0181	.0179	.....	.0175	.....	.0173	.....	.0173	.0173	.....	.....	.....	.....	" " " " " "

* Computed from data given in hydraulics by Hughes and Safford, pp 319 to 329, 1911.

From which

$$\frac{h}{3} = h_3 \tag{220}$$

That is, the maximum power from any pipe line will result when the friction head is one-third of the total head.

263. Economical Size of Steel Pipe Lines.—The choice of an economical size of pipe line becomes a very important matter where this pipe must be as long as in the high head plants of the west and some low head plants in which its first cost often becomes a considerable per cent. of the total cost of the plant.

As the diameter of the pipe line is increased, its first cost increases; but a gain also results from a decrease in the power which is lost in pipe friction. The economical diameter is that for which the sum of the interest and depreciation on the first cost of pipe line and the annual value of the power lost in pipe friction will be a minimum.

From equation (216)

$$h' = \frac{f}{d} \frac{v^2}{2g} \tag{221}$$

$$q = av = \frac{\pi d^2}{4} v \tag{222}$$

Combining (221) and (222) there results

$$h' = \frac{8fq^2}{\pi^2 g d^5} \tag{223}$$

From previous equation

$$P' = \frac{qh'}{8.8} \tag{2}$$

Then combining (223) and (2)

$$P' = 0.00287 \frac{fq^3}{d^5} \tag{224}$$

From principles of mechanics

$$t = \frac{62.5hd}{2S \times 12} = 2.6 \frac{hd}{S} \tag{225}$$

Now since a cubic foot of steel weighs 490 pounds

$$C = \frac{490 \, \pi dtc}{12} = \frac{chd^2}{S} \tag{226}$$

$$I = 333 \frac{chid^2}{2} \tag{227}$$

From (224)

$$B = 0.00287 \frac{fbq^3}{d^5} \tag{228}$$

From which

$$L = I + B = 333 \frac{chid^2}{S} + 0.00287 \frac{fbq^3}{d^5} \tag{229}$$

The economical diameter of pipe is that which will make L a minimum or

$$\frac{dL}{d\,(d)} = \frac{666\ chid}{S} - 0.01435 \frac{fbq^3}{d^6} = 0 \tag{230}$$

From which

$$d^7 = 0.0000\,2154 \frac{fbSq^3}{cih} \tag{231}$$

or

$$d = .2153 \sqrt[7]{\frac{fbSq^3}{cih}} \tag{232}$$

Since f is itself a variable, depending on both d and v, this formula is not exact. The formula of Darcy, or some other well known expression for f might be used in equation (229) but the solution of the resulting equation for d then becomes very much different and gives values differing but slightly from equation (232).

The economical diameter as given by equation (232) decreases as the head increases. In practice, the pipe line should be divided at convenient points into sections of equal diameters and the value of h to be used in equation (232) for finding the diameter of any section should be the average head of that section of pipe, since h enters in equation (229) only to the first degree.

c, i, q and S can be estimated as in any other problem of design. Liberal allowance for water hammer and efficiency of riveted joints should be made in choosing S. An average value,—say .018 of f should be chosen, and the values of d for each section of pipe computed. v can now be found, since $q = \frac{\pi d^2}{4} v$ and together with the approximate value of d will give the arguments for a close estimate of f. Equation (232) can then be applied for the second time and the resulting diameters will be fully as accurate as the data.

b must first be assumed. After the first preliminary determination of d, the cost of the pipe line can be found and a closer value for b determined.

264. Economy in Pipe Lines With Low Heads.—The economical diameter of pipe lines as determined by formula (232) applies only when the penstock can be made of the correct theoretical thickness as given by equation (225).

Where the head is small, this can not be done, as it would often require a penstock as thin as one-sixth inch, which is impracticable. It is then customary to assume a minimum practicable thickness, say one-fourth inch $= t'$.

In this case the problem of economical diameter is entirely different.

B remains unchanged as in equation (228).

c is then obtained from the first part of equation (226) or

(233) $$C = 128.3dt'c \text{ and}$$

(234) $$I = 128.3dt'ci \text{ now}$$

(235) $$L = I + B = 128.3dt'ci + .00287\frac{fbq^3}{d^5}$$

Then for maximum economy

$$\frac{dL}{d\,(d)} = 128.3t'ci - .01435\frac{fbq^3}{d^6} = 0 \text{ from which}$$

(236) $$d^6 = .000112\frac{fbq^3}{t'ci}$$

(237) $$d = 0.2195\sqrt[6]{\frac{fbq^3}{t'ci}}$$

EXAMPLE.

Consider the penstock divided into three sections subjected to average heads of 80, 160 and 240 feet, respectively, as shown in Fig. 346, page 549.

Assume $f = .018$
$q = 100$
$b = 25$
$S = 8000$
$c = .05$
$i = .12$
$h = 80, 160 \text{ and } 240$
$t' = .25$

$$d_{80} = .2153\sqrt[7]{\frac{.018 \times 25 \times 8000 \times 100^3}{.05 \times .12 \times 80}}$$

$$= .2153\sqrt[7]{7{,}510{,}000{,}000} = 5.56 \text{ feet}$$

$v = 4.12$ feet per second

$$d_{160} = 5.56\sqrt[7]{\tfrac{1}{2}} = 5.03 \text{ feet}$$

$$d_{240} = 5.56\sqrt[7]{\tfrac{1}{3}} = 4.75 \text{ feet}$$

From equation (225)

$$t_{180} = 2.6 \frac{hd}{S} = 2.6 \frac{80 \times 5.56}{8000} = .145 \text{ inches}$$

This is too thin for practical purposes and a new value .25 inches should be chosen.

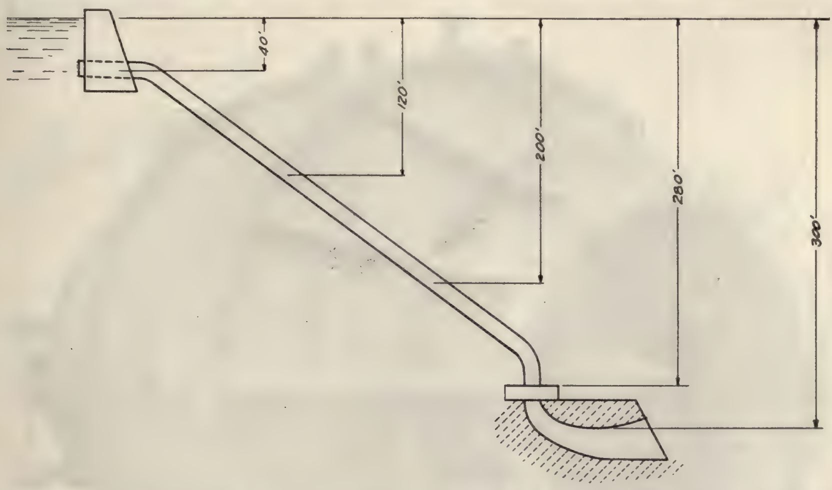

Fig. 346.—Illustrating the Application of the Method of Computation (see page 548).

Applying equation (237) with this value of t'

$$d = .2195 \sqrt[6]{\frac{fbq^3}{t'ci}} = 5.69 \text{ feet, say } 5.75 \text{ feet}$$

Investigating the second section in the same manner we have:

$$t_{160} = 2.6 \frac{160 \times 5.03}{8000} = .276$$

To assume the thickness at say $\frac{5}{16}$ inches and again apply the formulas would reduce the diameter but slightly. Therefore the economical dimensions will be practically as follows:

$$d_{160} = 5 \text{ feet}$$
$$t_{160} = \tfrac{5}{16} \text{ inches}$$
$$v = 5.1 \text{ feet per second}$$

For the third section $h = 240$ feet.

$$t_{240} = 2.6 \frac{240 \times 4.75}{8000} = .370 = \tfrac{3}{8} \text{ inches}$$

The theoretical economical diameters for the three sections were $d_{80} = 5.75$ feet, $d_{160} = 5.0$ feet and $d_{240} = 4.75$ feet.

It would hardly pay to involve the expense of reducers for these small changes in diameter, and the penstock should therefore be kept at about five feet diameter throughout, the exact diameter to be determined by weighing the three values of d according to the length of the respective sections.

Since the value of f depends upon the diameter, the assumed value of 0.018 should be corrected from published tables and the problem again solved.

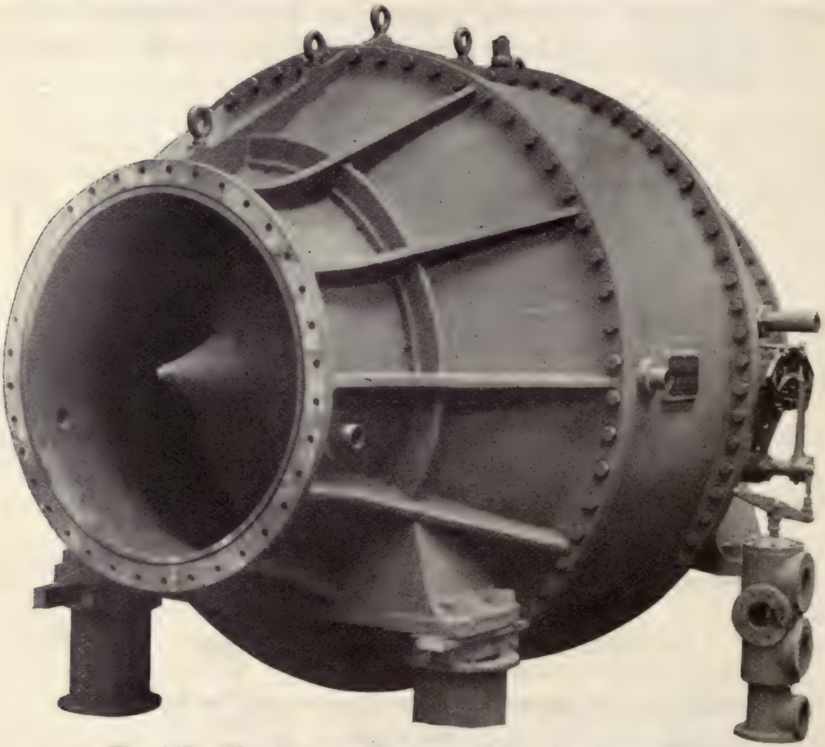

Fig. 347.—The Johnson Penstock Valve (see page 551).

265. Pipe Line Accessories.—The flow through the pipe lines is often controlled by gates at the headworks which close off the inlet (Fig. 364, page 572). In such cases, when the inflow is stopped, the water will usually drain from the pipe, producing a vaccum and resulting in a collapse of the line unless suitable air inlets are provided (see Figs. 361, page 568, 362, page 570, and 369, page 579). Air inlets of special design which will hold back the water but admit air before a vacuum occurs, should also be placed at the high points of long pipe lines.

When long pipe lines are used, the headworks are too far distant for convenient operation, and it is necessary to insert valves in the

lines or in the immediate vicinity of the power plant (Fig. 370, page page 580). Gate valves operated by hydraulic pressure or by an electric motor, are frequently used for this purpose. The Johnson valve (see Fig. 347, page 550, and Fig. 348) has proved very satisfactory for this service. This valve consists of a moving plunger of the needle type having two pressure chambers, one to close it, the other to open it. The valve

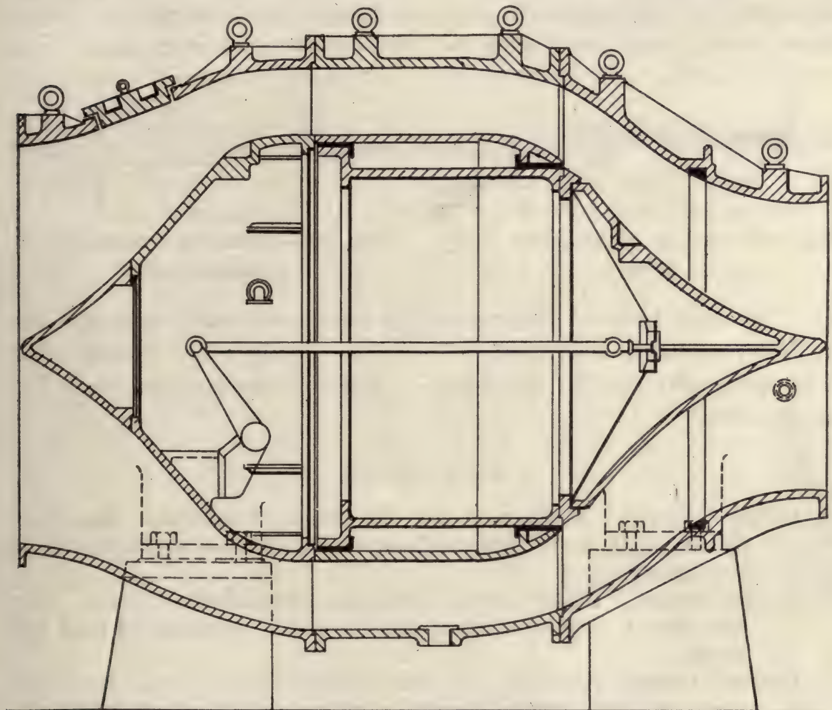

Fig. 348.—Longitudinal Section of the Johnson Penstock Valve.

is closed by admitting penstock pressure to the large chamber and exhausting it from the annular chamber to the atmosphere and is opened by reversal of this operation.

The supply and exhaust to and from the operating chambers are controlled by a valve of the balanced piston type, actuated by a floating lever, one end of which is attached to a lever on a rocker shaft which is connected internally to the moving plunger. By means of this arrangement, the plunger cannot travel faster than the control valve, and if it should be desired to stop the stroke of the valve plunger at any intermediate position, it can readily be done, and the

plunger will be automatically maintained at that position regardless of leakage either around the plunger or through the control valve.

The control valve, which is balanced and requires very little power, may be operated either by hand or by an electric motor, in which case hand control is also provided.

When the pipe line is of any considerable length, provision must be made for the expansion due to changes in temperature. Many form of slip joints have been used successfully (see Fig. 349). Fig.

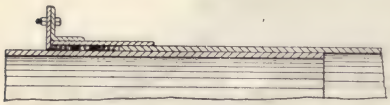

Fig. 349.—Detail of Penstock Slip Joint.

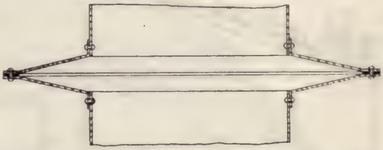

Fig. 350.—Detail of Diaphragm Expansion Joint.

350 * shows a form of diaphragm expansion joint used on a pipe line in Newfoundland where the temperature changes are considerable. Two sizes were used at this plant on pipes fifteen and seventeen feet in diameter.

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CHAPTER XVIII

EXAMPLES OF WATER POWER PLANTS

266. Sterling Plant.—A simple installation of three fifty-inch vertical Leffel wheels in the plant of the Sterling Gas and Electric Company of Sterling, Illinois, is shown by Figs. 351 to 354, pages 555 to 558, inclusive.

This plant is located on the Sterling race and is next to the last plant on the race on the Sterling side of Rock River (see Fig. 375, page 594). The head developed is about eight feet, and the power of each wheel is about 115 H. P. Each wheel is set in an independent wheel pit which can be closed by means of a gate, as shown in Fig. 354, page 558. The wheels are connected to a common shaft by beveled gearings and the general type of harness used is fully shown in the plan and elevation and needs no further description. In order to make repairs on any wheel without interfering with the other wheels, the wheels and harness are well supported from the foundation, a very essential condition for permanently maintaining a high efficiency. The discharge pit is of ample size, so that the velocity with which the escaping waters leave the draft tube is reduced to a practical minimum. A rack, to keep coarse floating material from the wheel, is placed in front of the penstock and is shown in Fig. 353, in section, and in Fig. 354, in partial elevation. The shaft from this plant is extended into the adjacent building and to it are belted the generators which supply electric current for light and power purposes in the city of Sterling. An engine is also connected to this main shaft, which is utilized in case of extreme low water conditions, where sufficient water for power is not available, or for flood conditions where the head is practically destroyed.

267. Plant of York-Haven Water Power Company.—Figure 355, page 559, shows the arrangement of the power station of the York-Haven Water Power Company on the Susquehanna River at York, Pa.

The power house is 478 feet long and fifty-one feet wide. The head-race is 500 feet long and of an average depth of twenty feet. The wheel pits are nineteen feet deep and extend the entire width

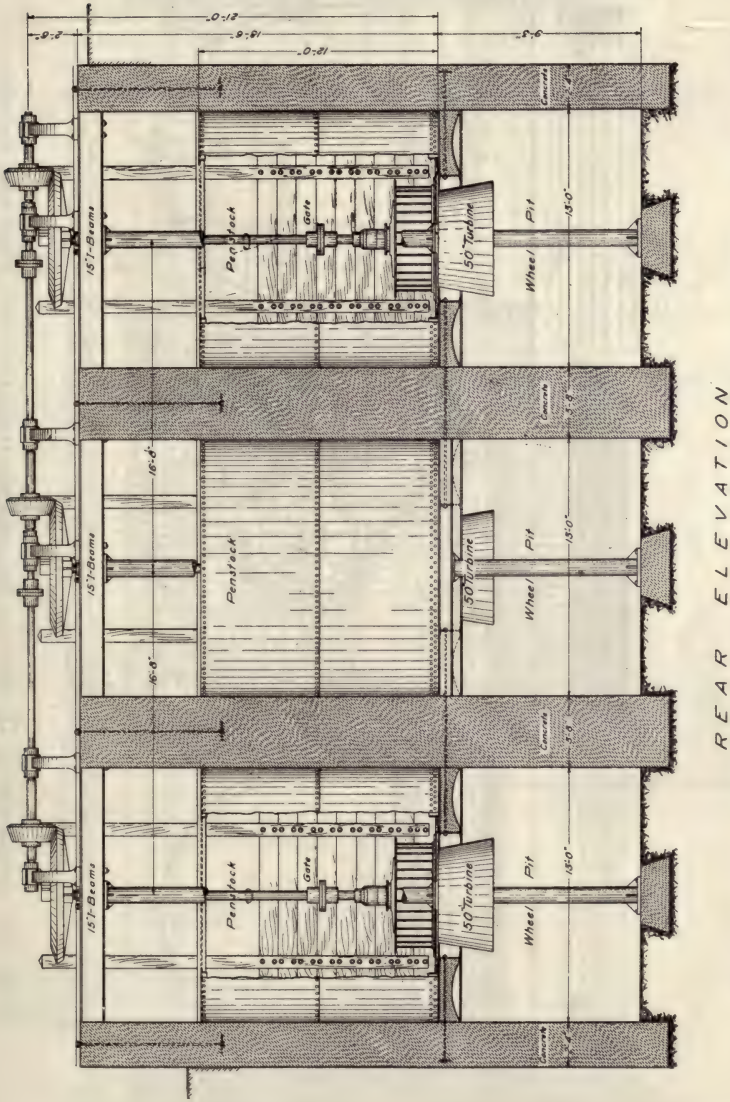
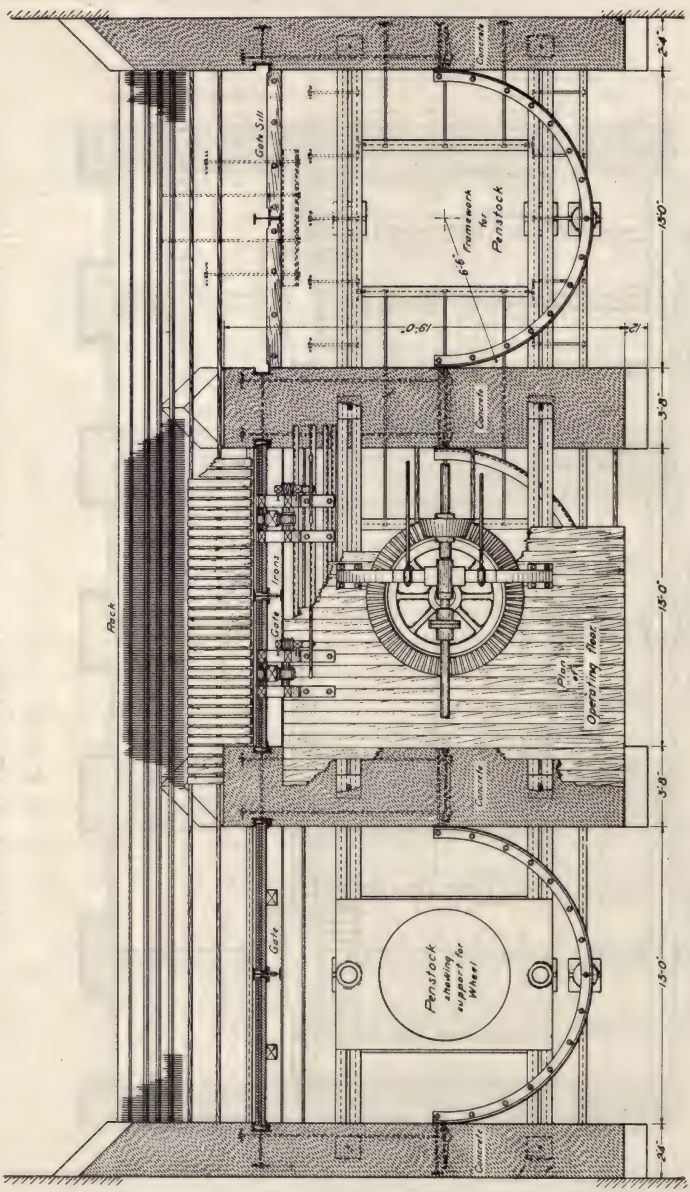


Fig. 351.—Wheel Pits at the Sterling Gas and Electric Light Co.'s Plant (see page 554).



PLAN

Fig. 352.—Wheel Pits at the Sterling Gas and Electric Light Co.'s Plant (see page 554).

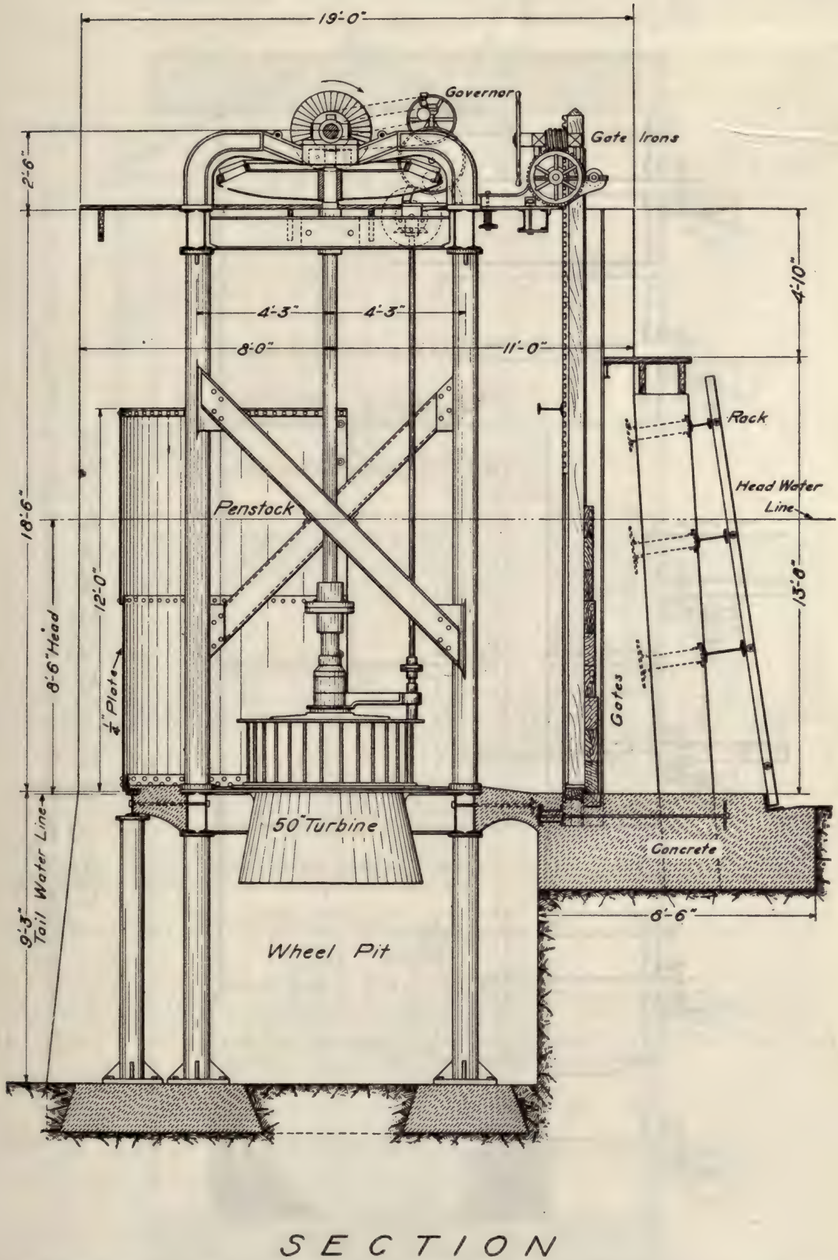
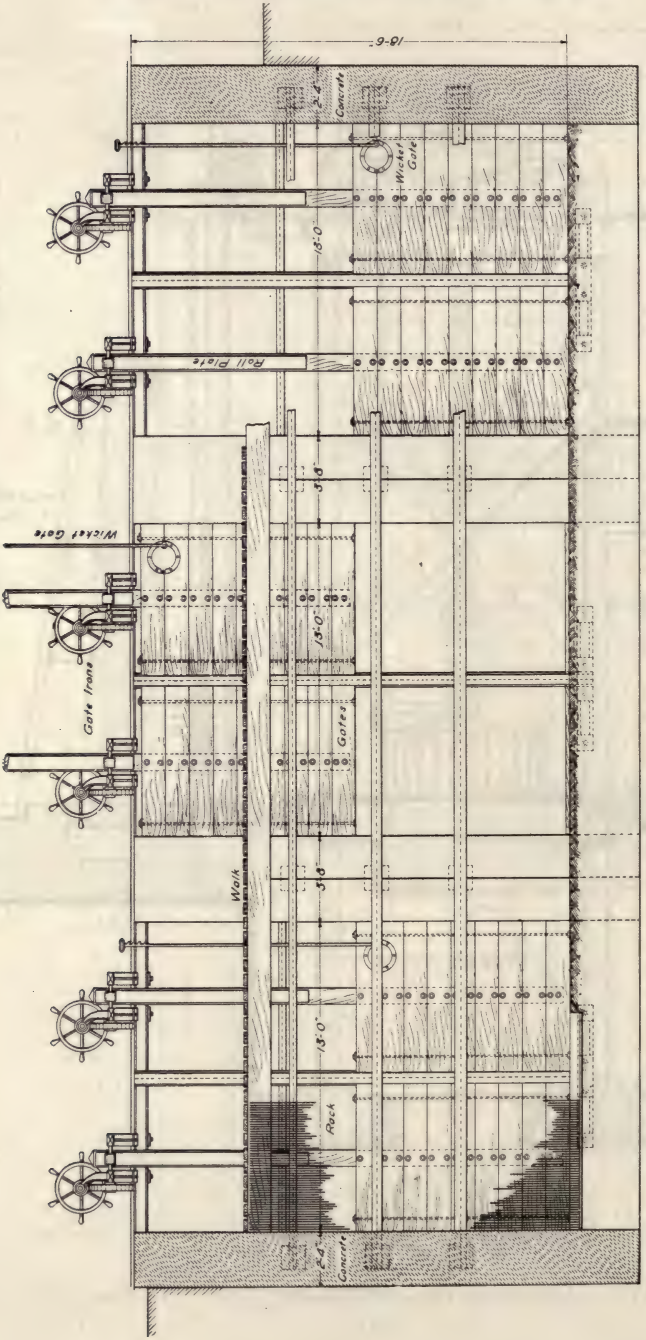


Fig. 353.—Wheel Pit, Sterling Gas and Electric Light Co.'s Plant (see page 554).



FRONT ELEVATION

Fig. 354.—Wheel Pits at the Sterling Gas and Electric Light Co.'s Plant (see page 554).

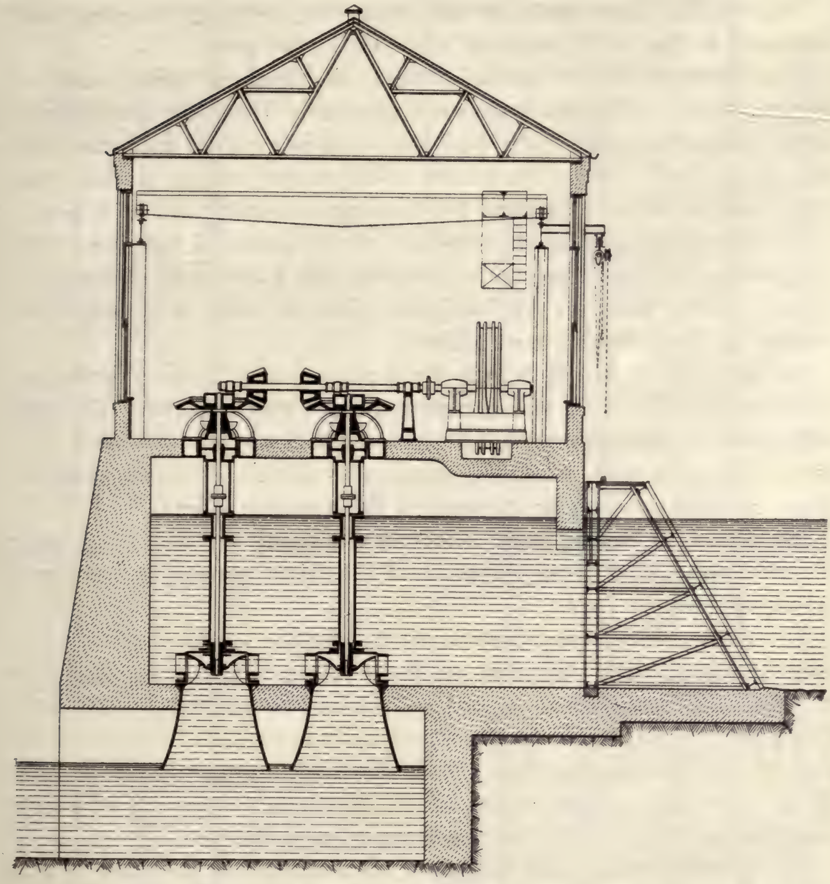


Fig. 355.—Plant of York Haven Water Power Co. (Electrical Engines). (See page 554).

of the power house, opening to the forebay. They are protected by iron racks and are made accessible by large head-gates of structural iron which weigh about eleven tons each.

Each pit contains two 78.5 inch flow turbines, hung from spring bearings just above the runners. The turbines are set on the floor of the pit and are about six feet above the lower water mark.

The draft tubes are ten feet long and extend well under water. The net head under normal conditions is about twenty-one feet. Float gauges on the switch board show at a glance the height of head and tail water.

The turbines were built by the Poole Engineering Company, and are rated at 550 H. P. each, or 1,100 H. P. per pair.

The turbines are of special design, the buckets being made of pressed steel. The shaft extends vertically from the turbines to bevel gears above the main floor and each is encased in a cast iron tube to protect it from the action of the water and to secure longevity both to the shaft and to the bearings which retain it in line.

The present installation consists of ten pairs of turbines with ten generators, equipped with Sturgess and Lombard governors.

The turbine bearings are supplied with oil from a gravity tank located on the switch-board gallery.

The generators are S. K. C., three-phase, sixty cycle alternators, rated at 875 kilowatts, and generate a 2,400 volt current. The normal speed of the generators is 200 revolutions per minute. Two 250 K. W., 125 volt, S. K. C., compound-wound, direct-current exciters furnish the exciter current to the generator fields.*

268. Plant of South Bend Electric Company.—Figure 356, page 561, shows the plant of the South Bend Electric Company at Buchanan, Michigan, built in 1901.

The dam, which was constructed in 1895, is of the gravity type, built of wood, with two rows of sheet piling below and one above it. It is about 400 feet long, and affords an average head of ten feet. The power is estimated to reach a minimum of 2,000 H.P. for from four to six weeks in a year, while the maximum will reach 5,000 H. P. On an average, 2,500 H. P. is available for about three months and 4,000 H. P. for the remainder of the year.

The power house, placed a short distance below the dam, is 273 feet long and forty feet wide. It is built of stone, with concrete foundations, and slate roof. It parallels the river so that the water from the turbines is discharged directly into the same. The regulating gates are seven in number, and are operated by racks and pinions.

The water wheels are Leffel turbines of sixty-eight inch vertical type, 300 H. P. each. They are geared to a line shaft, which extends nearly the whole length of the building, and to the end of which the generator is coupled. A forty inch vertical Leffel wheel is used for driving the exciter, which is belted to an intermediate shaft, driven by gears. The line shaft is divided into three units,

* See Electrical World, vol. 49, March 2nd, 1907.

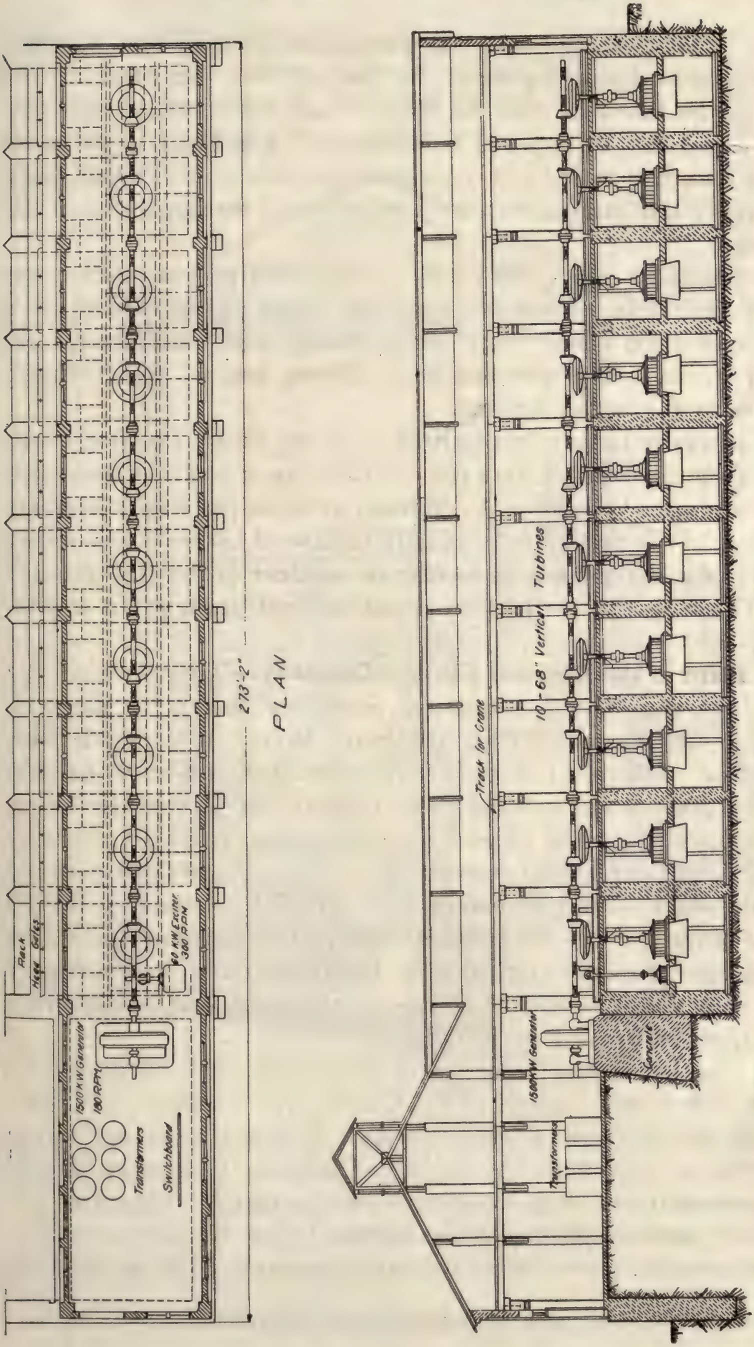


Fig. 356.—Plant of South Bend Electric Co., Buchanan, Mich. Vertical Turbine in Open Penstock Geared to Common Jack Shaft (see page 560).

so that either four, seven or ten wheels can be used for operating the generator, depending upon the load carried. In addition, the gears on the line shaft can be thrown out of mesh, so that any water wheel can be repaired if necessary. The plant is governed by two Lombard water wheel governors driven from the line shaft.

A twenty ton hand-operated crane serves all the apparatus in the building.

The generator is a 1,500 K. W., sixty cycle general electric revolving field type alternator supplying three-phase current at a pressure of 2,300 volts. The switch-board and transformers are located at one end of the building. There are no high tension switches at the power house.

The power is largely transmitted to South Bend, Indiana, a distance of sixteen miles, where the company has a steam power plant which is always kept in such condition as to be put into immediate operation. It is used, however, only in case of extreme low water, at times of a heavy peak, or in case of accident to the transmission line. The steam power house is used as a sub-station and distributing point.*

269. Plant of the Concord Electric Company.—This plant, shown in Fig. 357, page 563, is situated at Sewall's Falls on the Merrimac River, about four miles from the State House in Concord, New Hampshire. The dam is a timber crib-work structure about 500 feet long and gives a fall varying from sixteen feet to seventeen feet. The addition to the old plant is the one shown in cross-section by Fig. 357, and is of special interest due to the vertical shaft generating units which were here installed. Comparative estimates showed that all other features of the plant, except the machinery could be built cheaper with the vertical shaft installation and the machinery added only a few thousand dollars to the total cost, while other advantages determined its installation.

The new installation consists of two units, each consisting of three 55-inch bronze runners of the Francis type, mounted on a vertical shaft and hung on a step bearing. The machines are of the Escher-Wyss type built by the Allis-Chalmers Company, American representatives of the Escher-Wyss Company. The gates are of wicket pattern, controlled by Escher-Wyss mechanical governors, also built by the Allis-Chalmers Company. The generators,

* See Electrical World and Engineer, May 30, 1903, and July 14, 1906.

which are direct connected to the vertical shaft wheels, are of 500 K. W., three-phase, sixty cycle, 2,000 volt, 100 R. P. M., revolving field type. Excitation is furnished by one 75 H. P., three-phase, 2,600 volt induction motor, direct connected to a 45 K. W., 125

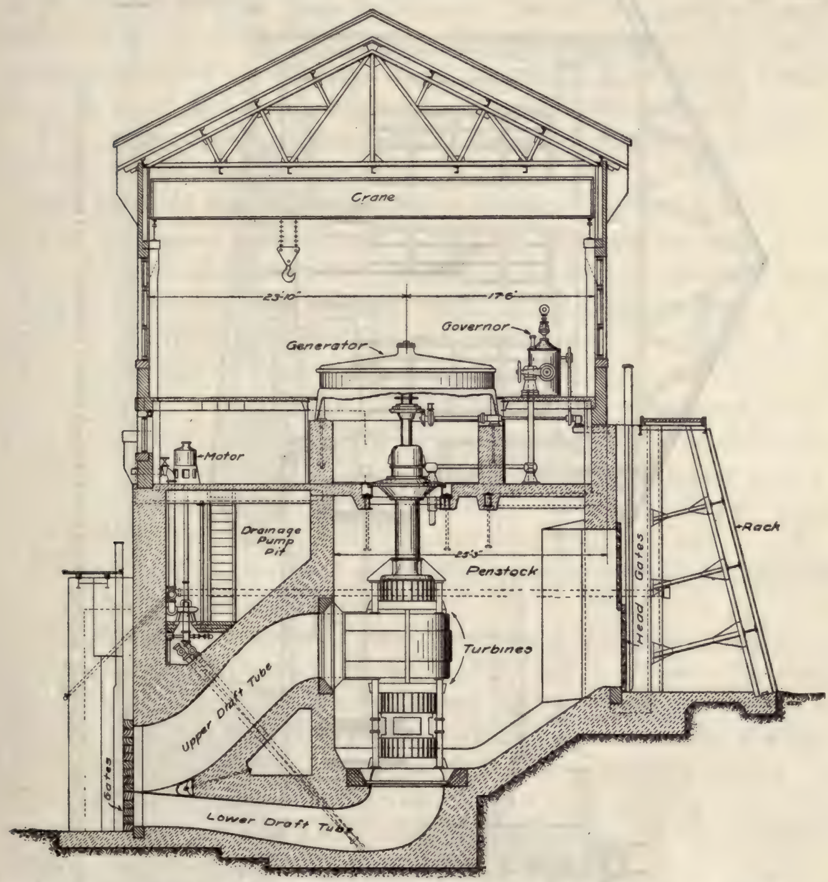


Fig. 357.—Plant of Concord Electric Co. Sewall's Falls Plant. Vertical Turbines Connected in Tandem (Engineering Records). (See page 562.)

volt, compound wound D. C. generator. The exciter unit runs at 680 R. P. M.*

270. **Plant of the Alpena Power Co.**—Figure 358, page 564, shows a section through the power house of the Alpena Power Company of Alpena, Michigan, on the Thunder Bay River. The head for this

* See Engineering Record, January 6th, 1906.

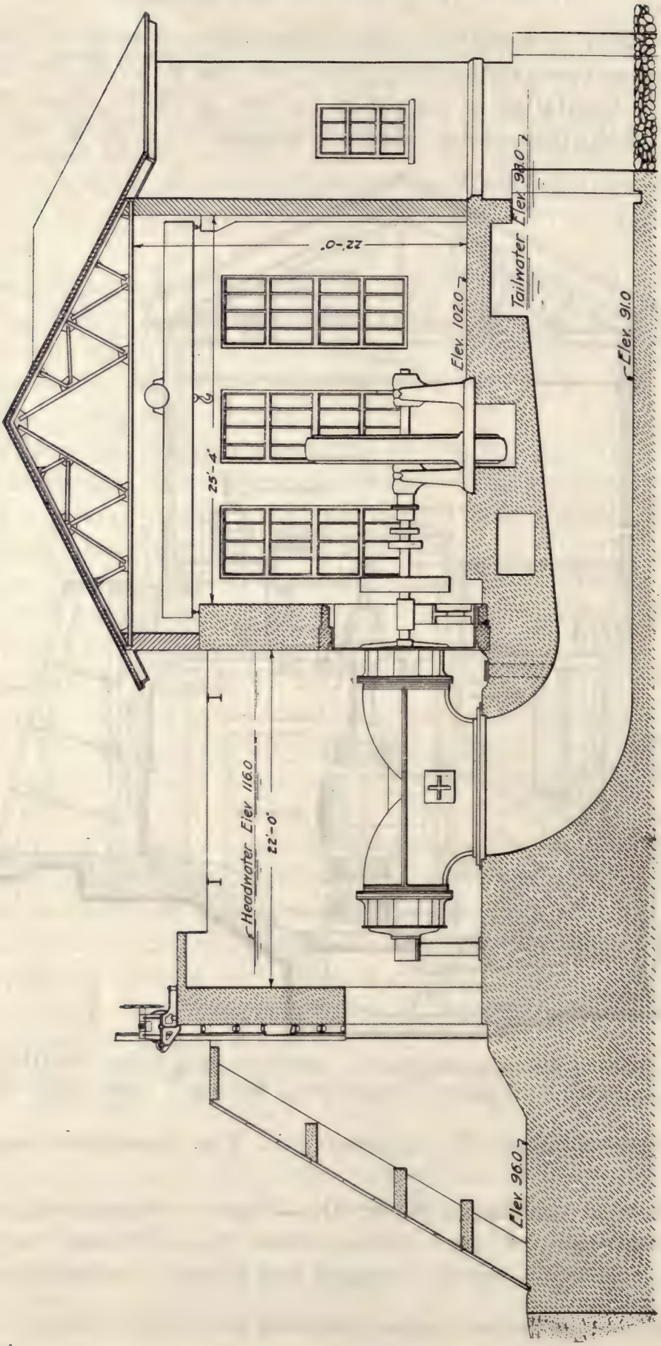


Fig. 358.—Section Through Power House of the Alpena Power Co. (see page 563).

plant is produced by a dam whose crest is twenty feet above low water. The dam proper is 186 feet long, 119 of which is tainter gate sections for flood discharge. The hydraulic works of the power station forms one end of the dam, the water being brought to the wheels in short open penstocks. The turbine units, of which there are three, are fifty inch twin Samson runners with horizontal shaft, designed to develop 600 H. P. at 120 R. P. M. under a head of sixteen feet and were made by the James Leffel Company.

The turbines are direct connected to 400 K. V. A. generators, which produce current at 2,300 volts.

271. **Plant of the Wisconsin River Power Company.**—The plant of the Wisconsin River Power Company, on the Wisconsin River at Prairie du Sac, Wisconsin, will operate under an average head of about thirty feet. Its ultimate capacity will be eight 2,500 K. W. units, each driven by four horizontal shaft tandem turbines. Each of the two pairs of twin turbines in each unit has a separate draft tube. The current generated is at 2,300 volts, part twenty-five and part sixty cycle, and is stepped up for transmission to 66,000 volts.

Figure 359, page 566, shows a cross-section of the power station. Each penstock is closed off by three head gates, and all head gates in the plant are manipulated by a traveling gantry crane.

The generators, turbines and turbine governors were furnished by the Allis-Chalmers Company.

272. **Plant of Columbus Power Company.**—The plant of the Columbus Power Company is shown in Fig. 360, page 567. It is situated on the Chattahoochie River just beyond the limits proper of the city of Columbus, Georgia, at a shoal known as Lovers' Leap. At this point a dam of cyclopean or boulder concrete with a cut stone spillway surface was erected giving a head of forty feet. The length of the dam is 975 feet eight inches, with a spillway 728 feet long.

Power house No. 1 is located at one end of the dam, so that no penstocks are necessary. Power house No. 2, which drives the plant of the Bibb Manufacturing Company, is supplied with pressure water by means of penstocks let through the bulk-head wall, which extends from house No. 1 to the river bank, power being transmitted to the mill by a rope drive system. In both cases the tail water is discharged into the excavated river bed beneath the power houses. Power house No. 1 is designed to develop 6,000 H. P. in six units, and No. 2 about 4,000 H. P. mainly in two units.

Power house No. 1 is 137 feet long and fifty-two feet wide. It rests on heavy stone foundations, the up-stream portions of which form the heavy bulk-head which is pierced by six large openings for plant No. 1, by a smaller opening for the exciter units and a larger one for the penstock leading to power house No. 2.

The openings for power house No. 1 are short flumes or chambers. The back end of each of the wheel chambers is closed with a

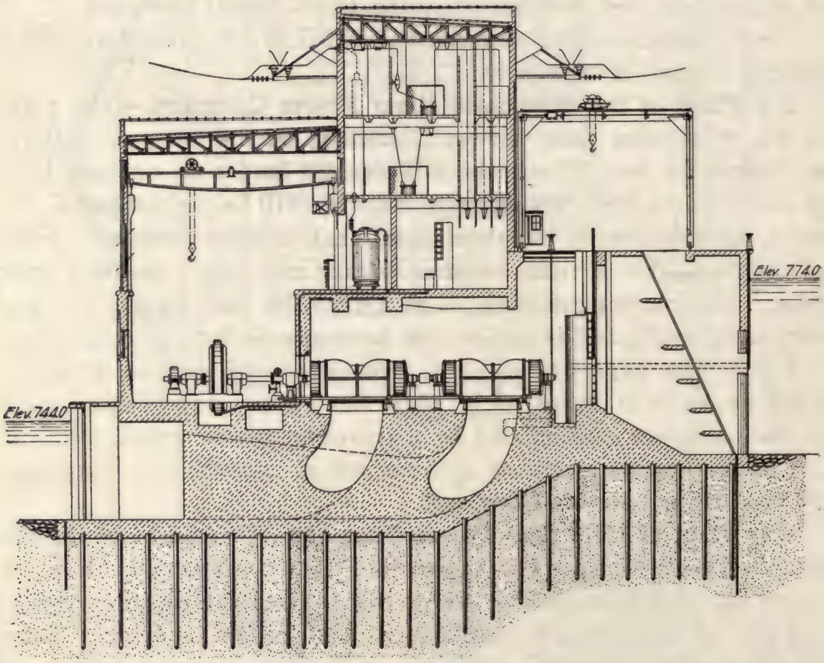


Fig. 359.—Cross-section of the Power House of the Wisconsin River Power Co. (see page 565).

heavy plate or bulkhead of cast iron and steel separating the wheel chamber from the generator room. The racks are of the usual construction and are supported on a framework of I-beams, giving them an inclination of about twelve degrees with the vertical. The gates to the wheel chambers are of timber and are raised by hand by means of a rack and pinion.

Each of the main wheel chambers contains a pair of horizontal thirty-nine inch Hercules turbines, which discharge into a common draft tube. The center line of the wheels is fifteen feet below normal head water level and twenty-five feet above normal tail water

level. Under the total head of forty feet, each pair of wheels develops 1,484 H. P. at 200 R. P. M. The draft tubes are seven and one-half feet in diameter at the turbine casing and ten feet at the discharge end.

Each pair of wheels is direct connected to a two-phase alternator built by the Stanley Electric Manufacturing Company. Each machine has a rated capacity of 1,080 K. W. at 6,000 volts and driven at 200 R. P. M. gives current at sixty cycles. Each is connected to the wheel shaft by a flexible leather coupling.

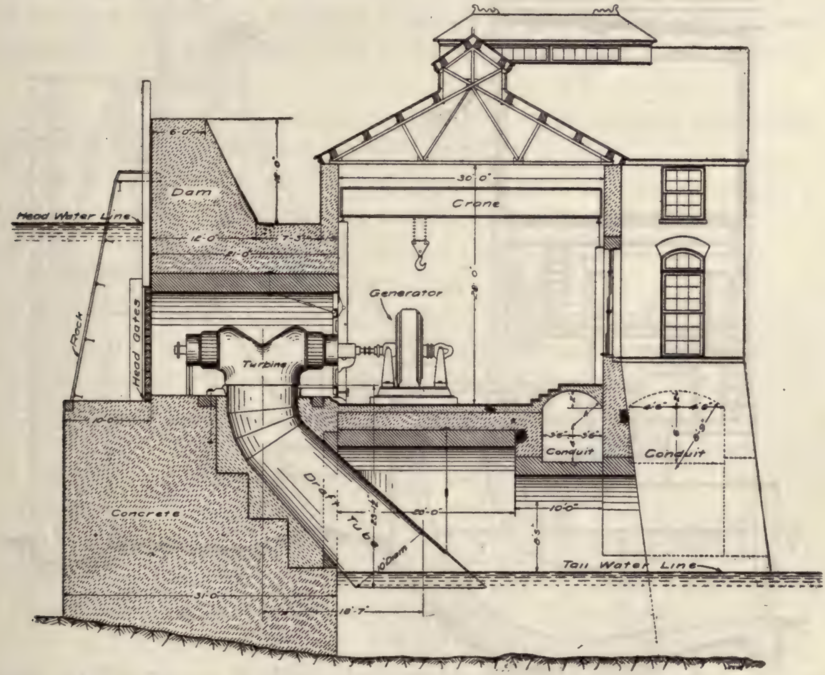


Fig. 360.—Plant of Columbus (Ga.) Power Co. Double Horizontal Turbines in Open Penstock (Engineering News). (See page 565.)

There are two exciters directly connected to a single eighteen inch Hercules wheel. Each exciter is of the Eddy type, having a capacity of sixty K. W. at seventy-five volts and running at 450 R. P. M. The exciters are under the control of mechanical governors.*

* Electrical World and Engineer, Jan. 23, 1904, or Eng. Record, Jan. 16, 1904.

273. Deerfield River Plant No. 2.—Figure 361 shows a vertical section of plant Number 2 of the hydro-electric development on the Deerfield River in Massachusetts. This is but one of several plants contemplated in the complete development and is located about three miles below Shelburne Falls. The head varies between fifty-three and sixty-three feet, with about fifty-eight feet as an average. The power house is located immediately downstream from the penstock headworks and is connected thereto by

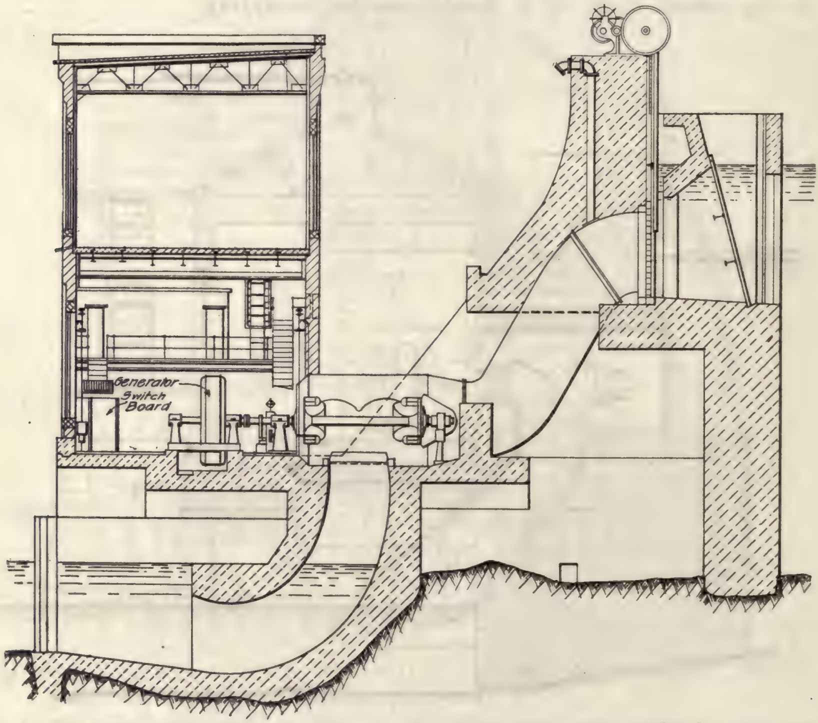


Fig. 361.—Section Through Deerfield River Plant No. 2.

three short steel plate penstocks, each eleven feet in diameter. The wheels are Wellman, Seaver, Morgan, horizontal double runner, central discharge turbines, which develop 3,200 H. P. per unit at 257 R. P. M. under fifty-eight foot head. The electrical installation is comprised of three 2,000 K. V. A., 2,300 volt, three-phase, sixty-cycle general electric generators, two motor driven exciters, two 2,300 K. W., three-phase, 2,300/6,600 volt, water cooled transformers*

*See Engineering Record, Feb. 1, 1913.

274. Plant of Winnipeg Electric Railway Co.—In Fig. 362, page 570, is shown the power plant of the Winnipeg Electric Railway Company. It is situated on the Winnipeg River at a point a few miles from Lac du Bonnet, which is on a branch line of the Canadian Pacific Railroad, sixty-five miles distant from the city of Winnipeg.

To obtain the necessary water, a canal 120 feet wide and with a clear depth of eight feet at normal low water was cut to the upper river near Otter Falls. The canal is eight miles long, with a drop of five feet to the mile, equalling a total head of forty feet. At the point where the dam is located there is a natural fall, and the dam crosses almost at the crest.

With the head and discharge available it is claimed that 30,000 electrical horse power can be developed.

The water wheels are all McCormick turbines made by the S. Morgan Smith Company, regulated by Lombard governors. The turbine pits are protected by racks to keep out ice, logs, etc.

The electrical units consist of four 1,000 K. W. and five 2,000 K. W. revolving field, sixty cycle, 2,300 volt, three-phase generators and two 100 K. W. 125 volt, direct-current exciters, all coupled to turbines, and two 175 K. W. 125 volt, direct-current exciters, coupled to three-phase 2,300 volt induction motors.

There are fifteen transformers, comprising five banks, by means of which the voltage is stepped up from 2,300 to 60,000 volts for transmission to the sub-station at Winnipeg over a distance of sixty-five miles.*

275. Spier Falls Plant of the Hudson River Power Transmission Company.—A cross-section of the Spier Falls Power house is shown in Fig. 363, page 571. A head of seventy-five feet, for operation of this plant, is derived from a granite rubble, ashlar-faced, masonry dam across the Hudson River between Mount McGregor and the Luzerne Mountains. The dam consists of 817 feet of spillway section, the remainder of the dam, 552 feet, being built about twelve feet higher. Water is admitted through arched gateways to a short intake canal designed to carry 6,000 cubic feet per second with a velocity of three feet per second. This canal distributes the water to ten twelve foot circular steel penstocks which lead about 150 feet to the wheels.

* See Electrical World, June 23, 1906.

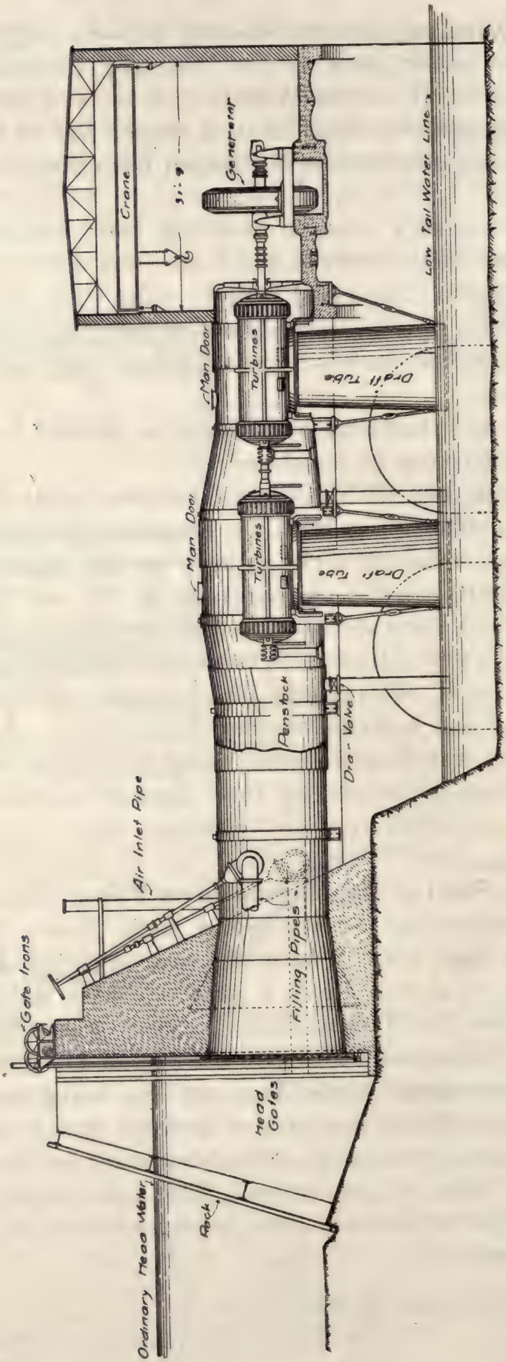


Fig. 362.—Plant of Winnipeg Electric Railway Co. Multiple Horizontal Turbines in Tandem in Steel Penstock (Electrical World, June 23, 1906). (See page 569.)

The power house is divided into three parts with the transformer and switchboard room in one end, the wheel room and generator room being formed by a longitudinal partition wall extending the length of the building, with traveling crane in each.

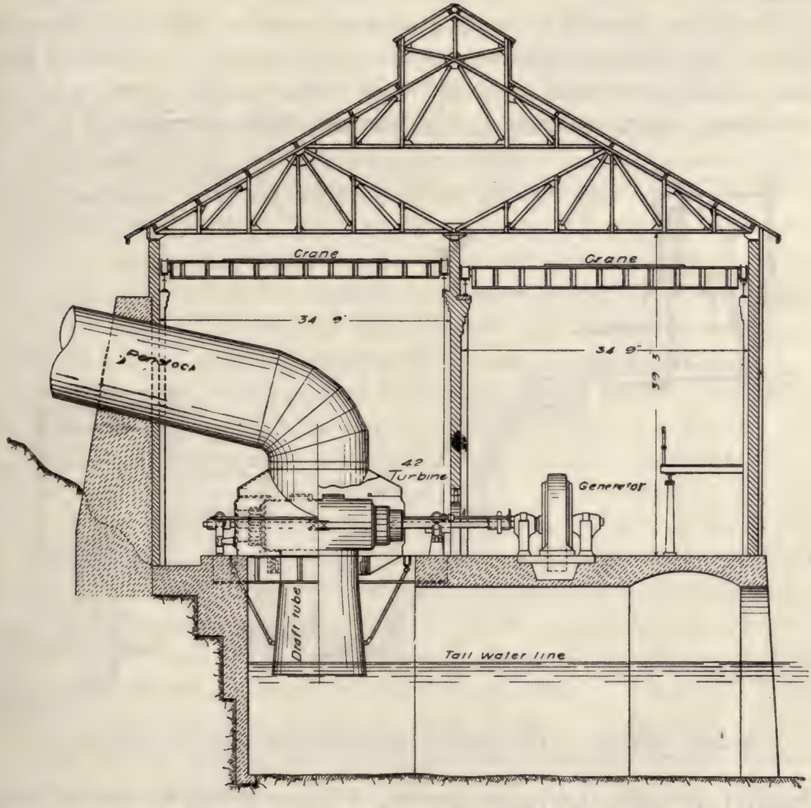


Fig. 363.—Plant of Hudson Water Power Co. Spier's Falls Plant. Double Horizontal Turbines in Steel Penstock. Central Discharge (Engineering Record). (See page 569.)

Each unit consists of a pair of forty-two inch or fifty-four inch cased S. Morgan Smith wheels, governed by Lombard and Sturgess governors and direct connected to 2,000 and 2,500 K. W. forty cycle, three-phase revolving field generators, built by the General Electric Company.

The transformer room contains seven 670 K. W. and thirty 833 K. W. General Electric Company air cooled transformers.

The power is distributed to Glens Falls, Schenectady, Saratoga Springs and Albany.*

276. Central Hydraulic Plant at Cohoes, New York.—Figure 364 shows a section of the central hydro-electric plant at Cohoes, New York.

This plant is built to replace several separate plants at the same place. The first installation amounts to 30,000 H. P., which is generated and distributed at 12,000 volts and will all be used within a relatively short distance. The development consists of a dam,

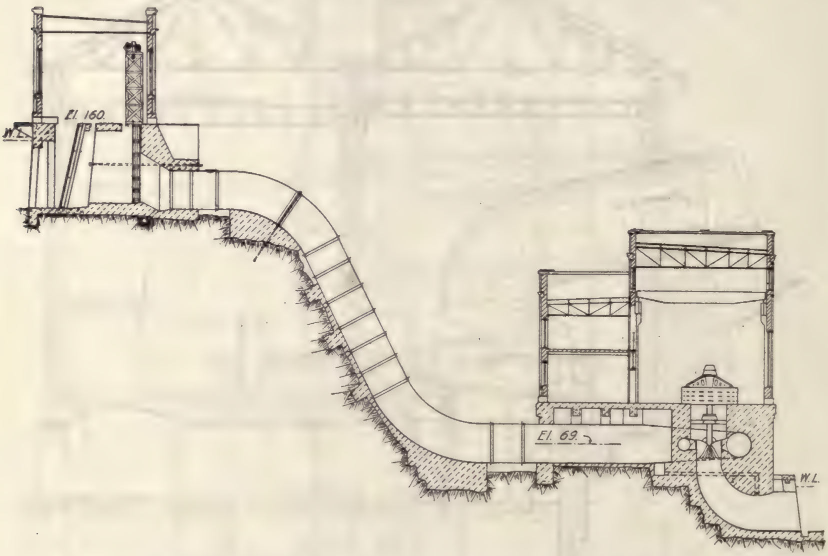


Fig. 364.—Section of the Central Hydro-Electric Plant at Cohoes, N. Y.

canal, penstocks and power house. The penstocks are five in number and eleven feet in diameter, each connected to a 10,000 H. P. vertical turbine with scroll case, made by the Platt Iron Works, which operates at 185 R. P. M. under a head varying from eighty-five feet during flood to ninety-eight feet during low water. The turbine runners are direct connected to the generators, which were manufactured by the General Electric Company, and are carried on a Kingsbury thrust bearing located above the generator.†

* Engineering Record, June 27, 1903.

† See Engineering Record, March 20 and March 27, 1915.

277. Plant of the Dolgeville Electric Light and Power Co.—In Fig. 365, page 574, is shown the plant of the Dolgeville Electric Light and Power Company at High Falls, New York, on what is now known as the Auskerada River.

The dam is built of limestone masonry. The height at the spillway is twenty feet, with each abutment six feet higher. The total length is about 195 feet. The width at the top is seven feet and at the bottom twenty-six feet. The upstream side is perpendicular, the downstream side being curved in order to properly receive and discharge the water. The head gate, twelve feet square and built in two sections, is fitted with a by-pass gate to relieve the pressure when filling the flume. The steel flume extends from the head gate to the power house, 520 feet away. This flume is ten feet in diameter, and is made of one-quarter inch steel plate, all longitudinal seams being double riveted. Just outside the dam is a vent pipe which assists in relieving the flume from any sudden strains.

There are two thirty-six inch horizontal Victor turbines made by the Platt Iron Works, each direct connected to one 450 K. W. 2,400 volt, two-phase Westinghouse generator. Each of these wheels will develop 600 H. P. at 300 R. P. M., under the working head of the water, which is seventy-two feet. They are mounted in cylindrical steel casings, and discharge downward through draft tubes, which extend a few inches below the surface of the tail water. Each wheel is supplied with a Giesler electro-mechanical governor.*

278. Great Falls Plant on the Passaic River.—Fig. 366, page 575, shows a sectional elevation of the development of the Great Falls of the Passaic River at Paterson, New Jersey. This plant, which is being carried out by the Society for Establishing Useful Manufacturing of Paterson, will supply 4,800 H. P. in the present installation and contemplates the addition of 1,700 H. P. The development will operate under a head of sixty-seven feet and will require a flow of 850 cubic feet per second, which will obtain for about 200 days per year. During low water periods the system will be supplied by an auxiliary steam plant of 6,000 H. P. capacity in a separate building at the site. The wheels are horizontal double runner turbines of the S. Morgan Smith Company, connected to 2,500 volt, sixty-cycle generators. Two of the main units are 1,720 H. P. each, and the third for first installation is 1,390 H. P., and

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* See American Electrician, April, 1898, Vol. 10, No. 4.

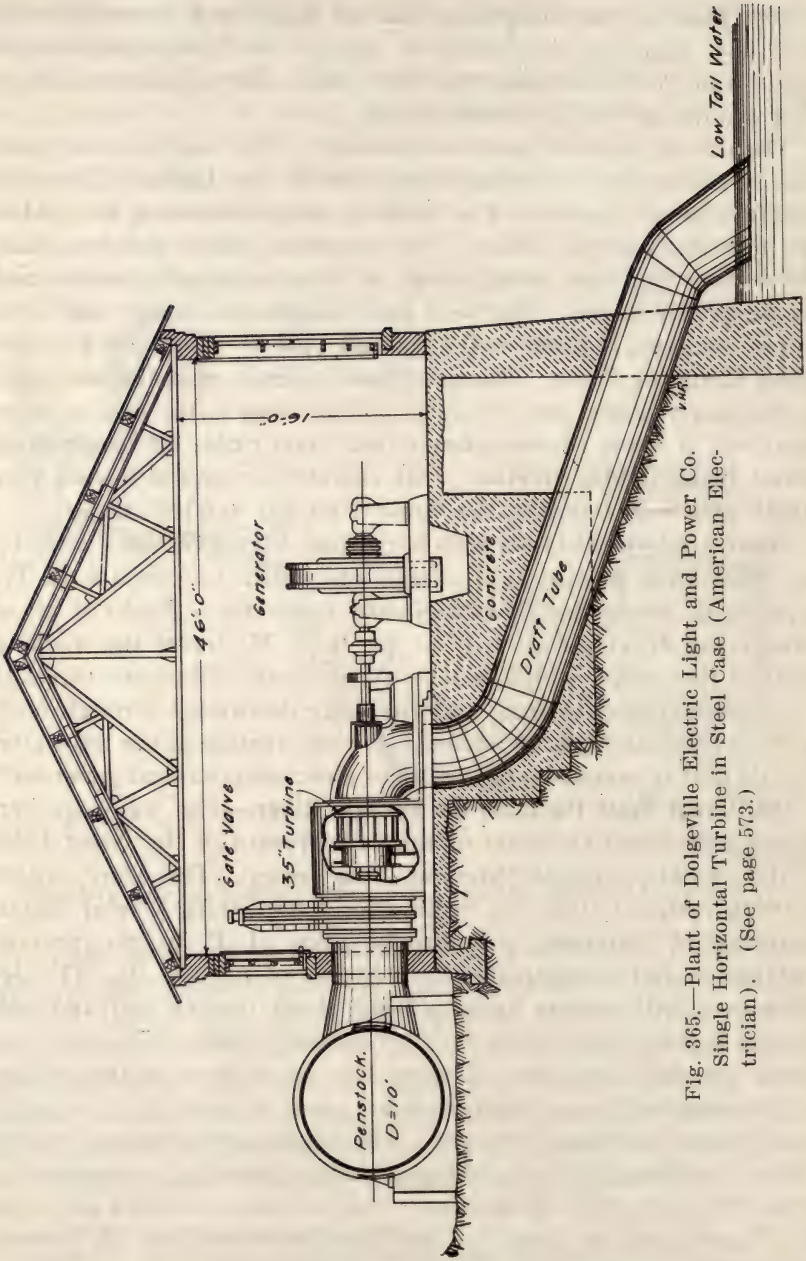

Fig. 365.—Plant of Dolgeville Electric Light and Power Co. Single Horizontal Turbine in Steel Case (American Electrician). (See page 573.)

provision is made for adding one more 1,720 H. P. unit. The exciter units are 100 H. P. each. The main turbine draft tubes are seventeen feet six inches long, six feet diameter at the top and eight feet diameter at the lower ends.*

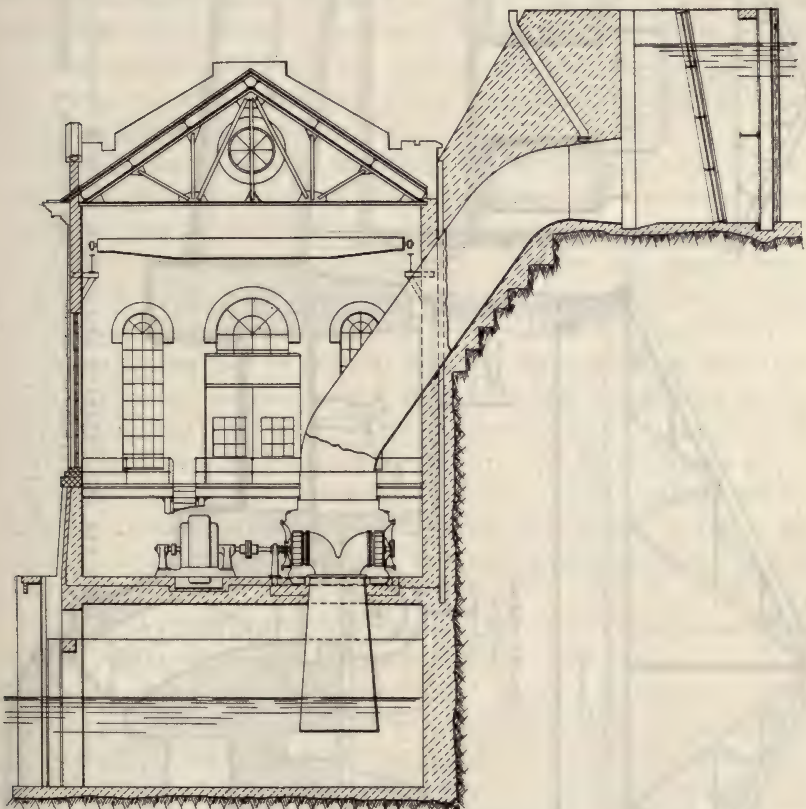

Fig. 366.—Sectional Elevation of Great Falls Development on the Passaic River (see page 573).

279. **Plant of the Shawinigan Water and Power Company.**—The power plant of the Shawinigan Water and Power Company is located on the St. Maurice River, Canada, at a point about twenty-one miles from Three Rivers, ninety miles from Quebec, and eighty-four miles from Montreal station. Fig. 367, page 576, shows a cross-section of their power station.

* See Engineering Record, Feb. 22, 1913.

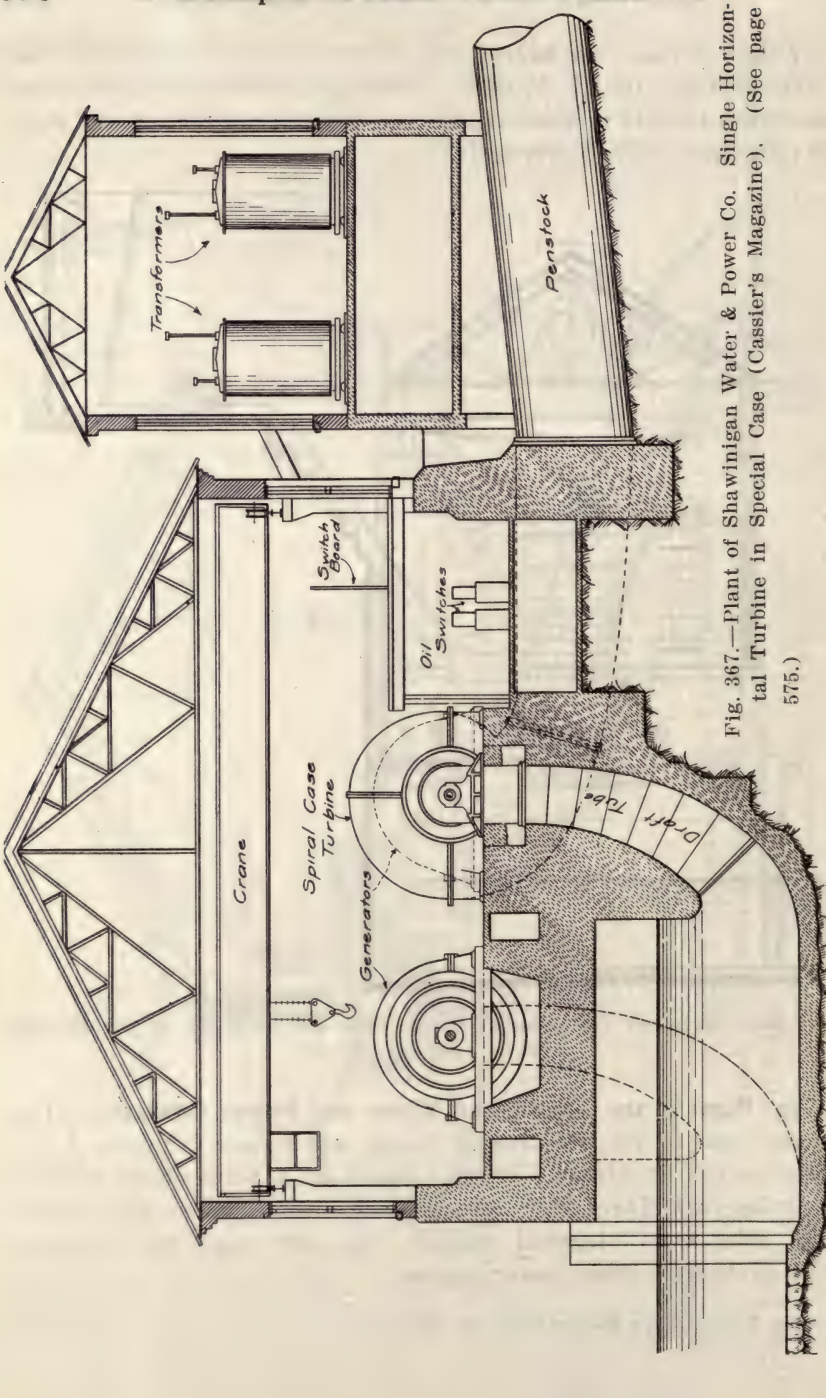

Fig. 367.—Plant of Shawinigan Water & Power Co. Single Horizontal Turbine in Special Case (Cassier's Magazine). (See page 575.)

The St. Maurice River has a total length of over 400 miles, and is supplied from a great many lakes and streams, the drainage area being about 18,000 square miles. The water flow is very steady throughout the year, and is in the neighborhood of 26,000 cubic feet per second, seldom going below 20,000 cubic feet per second. At the crest of the falls the water flows over a natural rock dam and then down over the cascade, making a fall of about 100 feet, then on in a narrow gorge through which the water rushes swiftly and in which there is a further fall of fifty feet.

The intake canal is 1,000 feet long, 100 feet wide and twenty feet deep. Its entrance from the river is located in a rather rapidly flowing stream at the crest of the falls where the water is twenty feet deep, for the reason that at times of rather high water, when the ice is flowing out of the river, the current is expected to carry the ice past the mouth of the canal. The end of the canal where it comes out at the face of the hill is closed by a concrete wall from which the water is led through steel penstock pipes down to the power house 130 feet below. The concrete wall or bulkhead in the canal is forty feet in height, about thirty feet in thickness at the bottom and twelve feet at the top. On top of this wall are set hydraulic cylinders for lifting the headgates and on top, covering the cylinders, is a brick gate house. The steel penstocks are nine feet in diameter.

The electrical apparatus was supplied by the Westinghouse Electric and Manufacturing Company and the turbines by the I. P. Morris Company.

The three turbine units of the original installation are horizontal double units of 6,000 H. P. These are direct connected with single 5,000 H. P. generator units of the rotating field type, with twenty poles. They are designed to operate at 180 R. P. M. giving two-phase currents at thirty cycles per second and 2,200 volts. A later installation consists of two 10,000 H. P. water wheels, each driving a 6,600 K. W. generator (see Fig. 159, page 249, and Fig. 236, page 366).

A separate penstock is provided for the exciter units which consist of two 400 H. P. turbines direct connected to exciters.*

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* See references as given: Eng. Rec., April 28, 1900; Can. Engr., April, 1901, May, 1901, and May, 1902; El. Wld. and Engr., Feb. 8, 1902; Cassier's Mag., June, 1904.

280. Wolf Creek Plant of the United Missouri River Power Co.—Figure 368 shows a vertical section of the development of the United Missouri River Power Company, known as the Wolf Creek plant. The hydraulic works of the power house are designed as a part of the main dam structure as shown. The wheels were made by S. Morgan Smith Company, each unit consisting of two bronze runners on a single horizontal shaft and discharging into a draft tube placed between the wheels, and developing 8,800 H. P at full gate opening at the average head of 114 feet. The turbine units are direct connected to the generators, which produce current at 6,600 volts which is stepped up to 70,000 volts for transmission.*

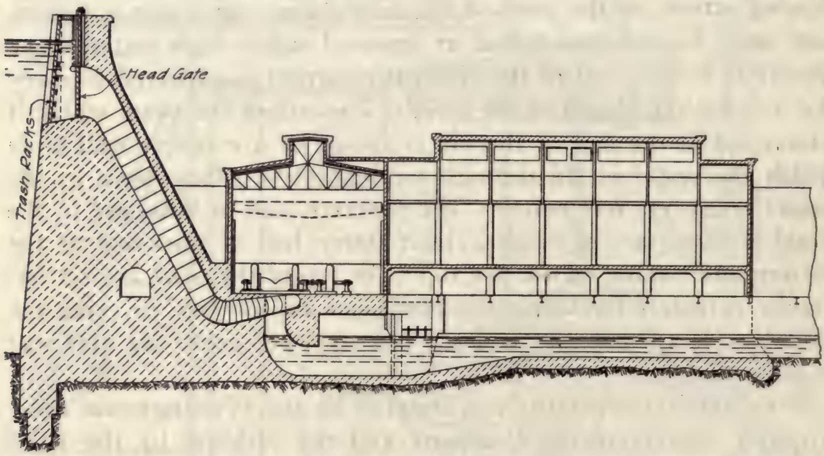

Fig. 368.—Section through the Wolf Creek Plant of the United Missouri River Power Co.

281. Ocmulgee Plant of Central Georgia Power Co.—Fig. 369, page 579, shows a vertical section of the plant of the Central Georgia Power Company, located between Atlanta and Macon on the Ocmulgee River. The plant will operate under a head of 100 feet, and the first installation will be 12,000 K. W., and the eventual capacity will be 18,000 K. W. The dam is of concrete with adjacent earthen embankments, the masonry portion being 1,070 feet long with a maximum height of 127 feet at the power house, which is located in the original stream bed.

The main turbine units each consists of a pair of thirty-nine inch McCormick runners on a horizontal shaft, rated at 5,500 H. P.

* Electrical World, Aug. 25, 1910.

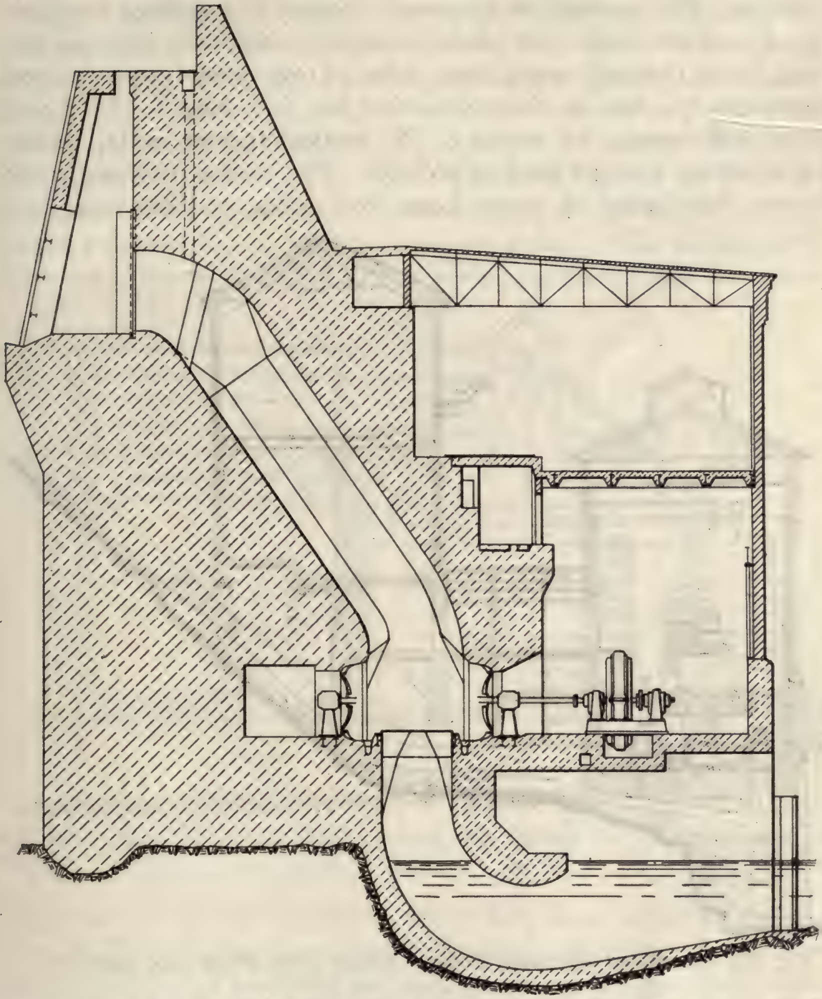

Fig. 369.—Vertical Section of the Ocmulgee Plant of the Central Georgia Power Co. (see page 578).

when operating at 300 R. P. M. under 100 foot head. Each unit is direct connected to 3,000 K. W., 2,300 volt, three-phase, sixty-cycle generator.*

282. **Tallulah Power Station.**—Fig. 370, page 580, shows a sectional elevation of power plant at Tallulah Falls near Atlanta,

* See Engineering Record, May 14, 1910.

Georgia. The ultimate development consists of a storage reservoir of 1,250,000,000 cubic feet capacity, an arch diversion dam 444 feet long, about one and one-quarter miles of rock tunnel, and six steel penstocks five feet in diameter and 1,200 feet long. The power house will contain six 10,000 K. W. vertical turbine units, operating under an average head of 600 feet. The turbine runners are of bronze, surrounded by a spiral cast iron casing, and discharge into

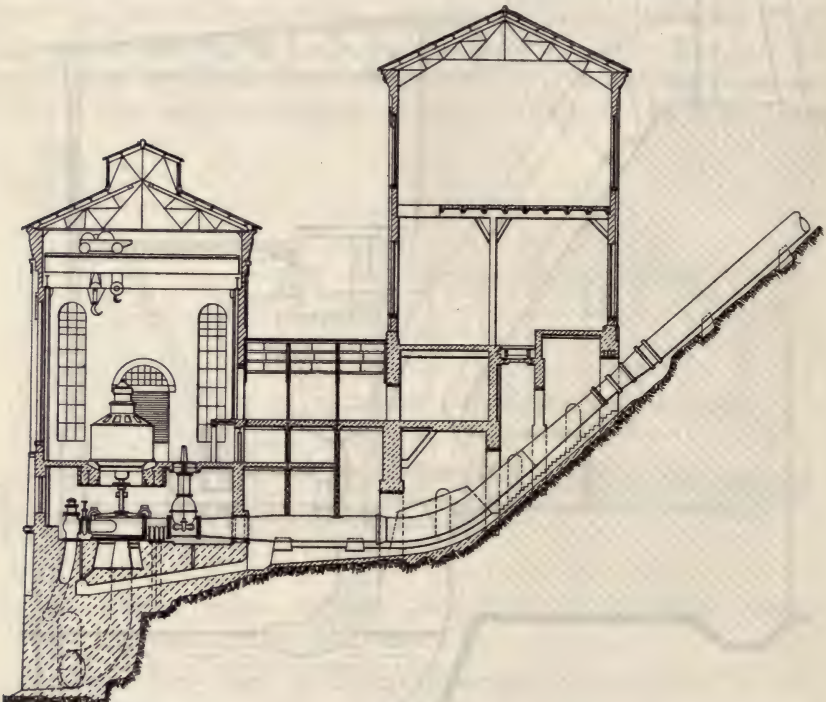

Fig. 370.—Sectional Elevation of the Tallulah Falls Plant (see page 579).

plate steel draft tubes which are concreted into the foundations. The generators are sixty-cycle, three-phase, 6,600 volt type, rated at 10,000 K. V. A. The exciters, carried on a stub shaft, extending upward from the main shafts, are 100 K. W., 250 volt machines, each designed to supply field excitation for two generators under full load.*

283. Plant of Nevada Power, Mining and Milling Co.—Fig. 371, page 581, shows a section through the plant of the Nevada Power

*See Water Power Chronicle, March, 1914.

Mining and Milling Company on Bishop Creek, near Bishop, Cal. The equipment of the station consists of two 750 K. W., sixty-cycle, 2,200 volt, three-phase alternating-current generators, running at 450 R. P. M. and a 1,500 K. W. generator running at 400 R. P. M. This latter generator is shown in the sectional drawing. There are two exciters of sixty K. W. each, delivering current at 140 volts pressure. Both exciters are operated by water wheels, and, in addition, one is provided with an induction motor. The water wheels were made by the Pelton Water Wheel Company of San Francisco.

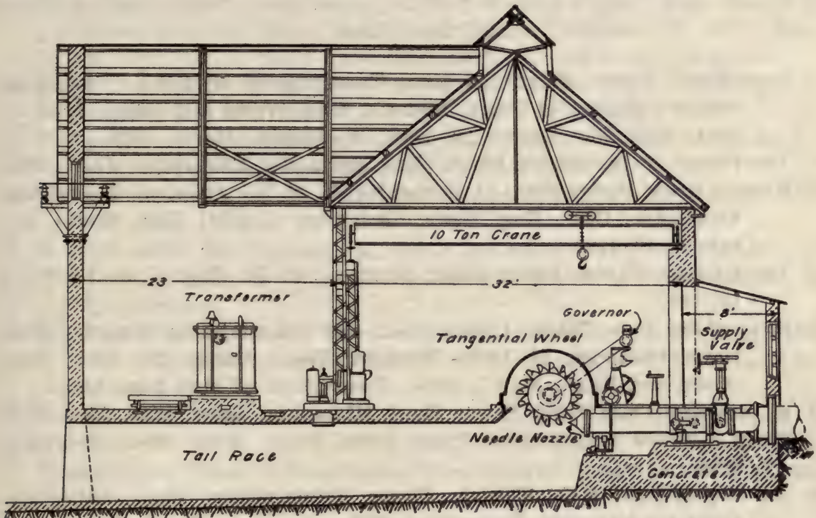

Fig. 371.—Plant of Nevada Power Mining and Milling Co. (Engineering Record).

The two 750 K. W. machines have Sturgess governors, and the 1,500 K. W. machine has a type Q Lombard governor. Hand control mechanism is provided for each wheel. Oil is supplied to the governor by two oil pumps operated by water wheels.

Water is taken from the creek at a small diverting dam and conveyed along the mountain-side in a pipe line. The pipe line is about 12,000 feet long, and consists of 6,700 feet of forty-two inch wood-stave pipe, 2,150 feet of thirty inch wood-stave pipe, and 3,150 feet of twenty-four inch steel pipe, all diameters being inside measurements. The forty-two inch pipe lies on a nearly level grade, the static head at the lower end being about thirty feet. At this point are placed two thirty inch gate valves, one opening into the thirty

inch pipe and the other provided for a future line. The thirty inch pipe descends the hill to a point that gives a static head of 265 feet. Here it joins the twenty-four inch steel pipe, which descends a steep hill to the power house, the total static head being 1,068 feet.

The power generated at the plant is transmitted, over a line of stranded aluminum, equivalent to No. 0 copper, to Tonopah and Goldfield, Nevada, making a total length of line of 113 miles. In crossing the White Mountains the line reaches an elevation of over 10,500 feet.*

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14. The Northern California Power Company's Systems. Elec. World and Engr., September 10, 1904.
15. Hydraulic Power Plant on the Puyallup River near Tacoma. Eng. Rec., October 1, 1904; Eng. News, September 29, 1904; Elec. World and Engr., October 1, 1904. Describes plant to furnish power to Seattle and Tacoma.
16. Hydro-Electric Developments on the Catawba River, South Carolina. Elec. World, May 25, 1907; Eng. Rec., July 30, 1904; Elec. World and Engr., July 23, 1904. Describes plant near Rock Hill, S. C.
17. Atlanta Water and Electric Power Company's Plant at Morgan Falls, Ga. Eng. Rec., April 23, 1904. Describes 14,000 H. P. plant.
18. New Water Power Development at New Milford, Conn. Eng. Rec., February 13, 1904; Elec. World and Engr., February 13, 1904; Elec. World, June 2, 1910. Describes plant on Housatonic River.
19. Generating System of the Portland General Electric Company. Eng. Rec., August 12, 1905. Description of 30,000 H. P. plant.
20. Water Power Development at Little Falls, Minn., and Its Industrial Results. Eng. Rec., June 13, 1905.
21. The Hydraulic Development of the Sterling Hydraulic Company. Eng. Rec., Dec. 16, 1905.
22. Hydro-Electric Development at North Mountain, Cal. Elec. World and Engr., March 4, 1905. Describes twenty-five cycle installation in California.
23. Water and Electric Power Plant of the Utah Sugar Company. Eng. News, April 13, 1905. Describes plant built for irrigation but available for power.
24. Plant at Rock Island Arsenal, Rock Island, Ill. West. Elec., November 23, 1901.
25. Transmission Plant of Kalamazoo Valley Electric Company, Mich. Amer. Electrician, July, 1901.
26. Plant of the Michigan-Lake Superior Power Company, Sault Ste. Marie. Amer. Electrician, August, 1898; Eng. News, September 25, 1902; Elec. World and Engr., November 8, 1902; Eng. Rec., November 2, 1907. Describes low head plant.
27. Power Transmission in Pike's Peak Region. Elec. World and Engr., July 26, 1902; Elec. World, May 26, 1906; Eng. Rec., May 19, 1906; Eng. Rec., July 19, 1902; Eng. News, January 1, 1903. Description of unusual plant of 1,600 K. W. capacity.
28. Joliet Water Power of Chicago Drainage Canal. Eng. Rec., April 19, 1902.
29. Chaudiere Falls Power Transmission Company, Quebec. Elec. World and Engr., June 15, 1901; Eng. News, May 7, 1903. Description of developments.
30. Plant of the South Bend Electric Company, South Bend, Ind. Elec. World and Engr., May 30, 1903.
31. Garvin's Falls Plant, Manchester, N. H. Eng. Rec., January 24, 1903; Eng. News, March 19, 1903; Elec. World and Engr., May 28, 1904; Assn. Eng. Soc., October, 1905. Description of 3,950 K. W. plant on Merrimac River.

32. Hudson River Power Plant at Spier Falls, N. Y. Eng. Rec., March 8, 1902; Eng. Rec., June 27, 1903; Elec. Review, July 21, 1906. Describes plant developing 20,000 H. P.
33. Puget Sound Power Company's Plant. Elec. World and Engr., October 22, 1904; Eng. Rec., September 2, 1904; Proc. Am. Soc. C. E., May, 1905. Describes plant to furnish power to Seattle and Tacoma.
34. Development at Turner's Falls, Mass. Elec. World and Engr., August 12, 1905.
35. New Hydro-Electric Power Plant of Cornell University. Eng. Rec., May 20, 1905.
36. New Plant of Holyoke Power Company. Eng. Rec., September 15, 1906; Elec. World, September 15, 1906. Describes plant furnishing power to large paper mills.
37. The Hydraulic Power Development of the Animas Power and Water Company, April 14, 1906; Eng. Rec., Elec. Review, January 30, 1904; Eng. News, January 4, 1906. Describes high head plant in Colorado.
38. Construction of the Neals' Shoals Power Plant on Broad River, S. C. Eng. Rec., March 3, 1906. Describes plant and its construction.
39. Plant at Sewall's Falls, N. H. Eng. Rec., January 5, 1906.
40. The Warriors' Ridge Hydro-Electric Plant at Huntingdon, Pa. Eng. Rec., December 22, 1906. Elec. World and Engr., Dec. 22, 1906.
41. Plant of Vermont Marble Company, Proctor, Vt. Elec. World, February 3, 1906.
42. Hydraulic Developments at Trenton Falls, N. Y. Elec. World, May 19, 1906.
43. Hydro-Electric Power Plant and Transmission Lines of the North Georgia Electric Company. Elec. Review, October 20, 1906.
44. Low Head Hydro-Electric Developments in Michigan. Eng. Rec., October 19, 1907.
45. Greenville-Carolina Power Company, S. C. Elec. World, June 22, 1907; Eng. Rec., October 6, 1906. Describes low head plant.
46. Power Plants of the Portland Railway, Light and Power Company, Portland, Ore. Eng. News, June 27, 1907; Engineer, April 15, 1907.
47. Development of Electric Power at Shoshone Falls, Idaho. Electrician, March 9, 1907.
48. York Haven, Penn., Transmission Plant. Elec. World and Engr., September 19, 1903; Elec. World, March 2. 1907.
49. Tumwater Plant of the Great Northern Railway. Eng. Rec., October 30, 1909.
50. Southern Wisconsin Hydro-Electric Plant at Kilbourn, Wis. Elec. Rec., August 28, 1909; Eng. Rec., September 18, 1909. Description of 6,000 K. W. low head plant.
51. Juniata Water and Water Power Company. Elec. World, January 20, 1910. Description of hydraulic plant and steam auxiliary.
52. Capital City Plant of the United Missouri River Power Company. Eng. News, October 20, 1910. Description of plant to furnish power in western Montana.

53. Two Hydro-Electric Developments on the Peshtigo River. Eng. Rec., January 1, 1909; Elec. World, November 24, 1910; Eng. Rec., May 13, 1911. Describes plant at High Falls and Johnson's Falls, Wis.

54. Rainbow Falls Hydro-Electric Development. Eng. Rec., March 12, 1910; Cassier's Mag., July, 1911; Elec. World, July 6, 1912. Describes a plant generating 21,000 K. W.

55. Great Falls Power Plant of the Southern Power Company. Elec. Jour., December, 1907; Proc. Am. Inst. E. E., June, 1908; Elec. World, March 24, 1910; Elec. Jour., April, 1911; Elec. World, September 16, 1911; Elec. Rev. and West. Electrician, July 22, 1911. Describes development on the Catawba River.

56. Water Power Development of the Great Northern Power Company near Duluth, Minn. Elec. World, July 28, 1906; Eng. News, Dec. 26, 1907; Iron Age, January 2, 1908; Engrs. Soc. Univ. of Minn. Year Book, 1908; Power, August 11, 1908. Describes plant on St. Louis River.

57. The Colliersville Hydro-Electric Plant. Eng. Rec., March 7, 1908. Description of plant on Susquehanna River to supply railway.

58. Typical Low Head Hydro-Electric Power Plants. Trans. Eng. Soc. Univ. of Toronto, No. 20, 1908.

59. Power Development at Priest Rapids on the Columbia River. Eng. Rec., April 4, 1908; Elec. Rec. and West. Electrician, July 29, 1911. Describes extensive plant.

60. Sanitary District of Chicago. Development on Drainage Canal. Elec. Rec., February 8, 1908; Elec. World, January 12, 1911.

61. Sewall's Falls Plant, Concord, N. H. Power, May 25, 1909. Describes vertical multi-runner plant.

62. Development of the Connecticut River Power Company. Elec. World, September 9, 1909; Eng. Rec., March 27, 1909; Eng. Rec., April 3, 1909. Describes plant and transmission to several towns.

63. Hydro-Electric Plant of the Schenectady Power Company on the Hoosic River. Elec. Rec., March 27, 1909; Elec. World, May 20, 1909; Eng. Rec., July 24, 1909.

64. System of the Rochester Railway and Lighting Company. Elec. World, January 14, 1909, January 28, 1909. Describes plant with steam turbine auxiliary.

65. The McCall's Ferry Hydro-Electric Power Plant. Eng. Rec., September 21, 1907; Elec. Rec., June 1, 1907; Sci. Amer., March 13, 1909; Elec. Rec., February 27, 1909; Central Stations, June, 1909; Engng., January 7. 1910; Eng. Rec., May 28, 1910. Describes plant to furnish 135,000 H. P.

66. Central Georgia Power Company. Eng. Rec., April 17, 1909; Eng. Rec., May 14, 1910; Elec. World, April 27, 1911. Description of large development.

67. Hydro-Electric Generating Station of Great Western Power Company on the Feather River. Elec. World, August 26, 1909, and September 16, 1909; So. Electrician, February, 1913; Eng. Rec., June 11, 1910, and

July 16, 1910; Eng. Rec., June 11, 1910, and July 16, 1910; Proc. Brooklyn Engrs. Club, January, 1912. Describes plant to furnish power to Sacramento and surrounding country.

68. San Joaquin Light and Power Company. Eng. Rec., February 4, 1911, and February 11, 1911; Sci. Amer. Sup., July 6, 1912; Power, July 16, 1912. Describes a 16,000 K. W. plant.

69. Southern California Edison Company. Eng. Rec., March 18, 1905; Elec. World and Engr., March 11, 1905; Elec. Rev. and Electrician, March 25, 1911; Power, September 5, 1911. Describes stations at Redlands and Mill Creek.

70. Tennessee River Power Development. Eng. Rec., June 10, 1911; Elec. World, November 15, 1913; Proc. Brooklyn Engrs. Club, January, 1912; Elec. World, November 15, 1913; Elec. Rev. and West. Electrician, November 22, 1913. Describes largest project in the south.

71. Tallulah Falls Hydro-Electric Development of Georgia Railway and Power Company. So. Electrician, September, 1912; Elec. World, December 20, 1913; Eng. News, April 16 1914; Eng. Contng., May 6, 1914; Elec. Engng., May, 1914; Proc. Engrs. Club of Philadelphia, July, 1914; Power, January 27, 1914; Gen. Elec. Rec., June, 1914; Min. and Eng. World, October 3, 1914; Eng. Rec., March 21, 1914. Describes development under head of 600 feet.

72. Mississippi River Power Company Development at Keokuk, Iowa. Eng. News, September 28, 1911; Jour. W. S. E., March, 1912; Eng. Contng., May 29, 1912; Elec. World, May 31, 1913; Cassier's Mag., July, 1913; Power, August 5, 1913; Sci. Amer., September 13, 1913; So. Electrician, January, 1913; Cornell Civil Eng., February, 1913; Gen. Elec. Rev., February, 1914; Eng. (London), May 1, 1914. Three hundred thousand H. P. low head plant.

73. Skagway, Alaska, Hydro-Electric Central Station. Elec. World, October 21, 1911. Describes high head Pelton wheel installation.

74. White River Development of the Pacific Coast Power Company. Eng. Rec., April 13, 1912; Eng. News, April 11, 1912. Description of plant of 36,000 H. P. capacity.

75. Hydro-Electric Station at Gatun Spillway. Eng. Rec., July 27, 1912. Description of development.

76. Winnipeg Hydro-Electric Power Station. Elec. World, June 23, 1906; Electrician (London), March 1, 1912; Engng., July 26, 1912; Can. Elec. News, December, 1911. Describes the development.

77. Vancouver Island Power Company's Plant on the Jordan River. Elec. World, October 12, 1912; Eng. Rec., October 19, 1912; Can. Engr., November 28, 1912. Description of the development.

78. Butte Anaconda Hydro-Electric Power. Min. and Eng. World, November 30, 1912.

79. Deerfield River Power Developments. Eng. Rec., February 1, 1913.

80. Appalachian Hydro-Electric Developments on New River, Va. Power, March 25, 1913.

81. Horseshoe Falls Hydro-Electric Plant. Eng. Rec., February 7, 1914.
82. Coosa River Plant of the Alabama Power Company. Eng. Rec., April 4, 1914; Mfgrs. Rec., May 7, 1914; Proc. Amer. Soc. C. E., September, 1914. One hundred and five thousand H. P. low head plant.
83. Ocoee River Plant of the Tennessee Power Company. Eng. Rec., April 18, 1914; Mfgrs. Rec., April 2, 1914; Elec. Tract., November, 1913; Elec. Engng., February, 1914. Developments under head of 250 feet.
84. Salmon River Plant of Salmon River Power Company, N. Y. Elec. World, June 13, 1914; Eng. Rec., June 13, 1914. Description of high head plant.
85. Cedar Rapids Hydro-Electric Development. Eng. Rec., July 18, 1914, and July 25, 1914. Describes 160,000 H. P. low head plant.
86. Central Hydro-Electric Plant at Cohoes, N. Y. Eng. Rec., March 20 and March 27, 1915. Description of 50,000 H. P. development to replace several plants.

CHAPTER XIX

THE RELATION OF DAM AND POWER STATION

284. **General Consideration.**—In any water power plant the water must be taken from some source, conducted to the wheels, and discharged from the same at the lower head. To accomplish this object there must be a head-race flume or pipe line leading from the source of supply to the plant which may be of greater or less length and in which more or less of the available head may be lost in order to produce the velocity of flow and overcome the frictional resistance.

After entering the plant the water is conducted to the turbine T and discharged through a draft tube into a tail-race of greater or less extent in which there is also a loss caused by friction and

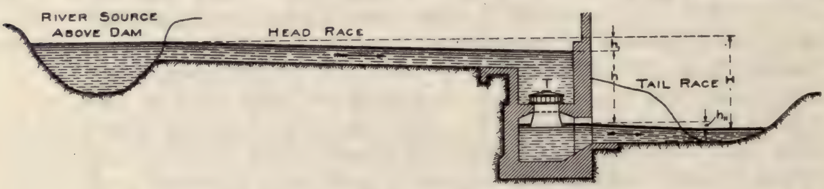

Fig. 372.—Relation of Power House to Reservoir.

velocity of flow, similar to that in the head-race. In Fig. 372, the total head available is H; the head lost in the head-race is indicated by h_1; and the head lost in the tail-race is indicated by h_2. The net energy of the wheel is $h = H - h_1 - h_2$, and a portion of h is also lost in the slip, leakage, and friction of the machinery and transmission.

The power plant should be located with reference to the dam so that (1) the greatest amount of head may be utilized at the least expense; (2) the plant constructed may be as free as possible from interruptions due to floods or other contingencies; (3) the location chosen may be at such a point where security of construction can be realized at minimum expense.

Each of these influences is of importance and the relative location of the power plant and dam must depend upon these and various other conditions which must be carefully considered.

285. Classification of Types of Development.—For the purpose of a clear understanding of the principles involved, the type of development may be grouped or classified into:

First: Concentrated fall, in which the plant is built in the dam or closely adjoining thereto, with a short or no raceways. In this case the entire fall is concentrated by means of the dam and as a rule this class of development is adaptable only to central power stations where one or two plants only are to be installed on the power.

Second: Diversion type with dam. In this case the fall is developed by means of a dam in the manner conforming to the last type but the water is distributed to one or more plants by means of a long head-race canal through which the water flows to the power station, after which it is discharged either into the stream of a long head-race canal through which the water flows to the power station, after which it is discharged either into the stream at some point below the dam or into a tail-race from which it is finally discharged at a point lower down the stream.

Third: Diversion with or without dam. In this case the development may be with or without a dam at the head of the rapids or fall which is to be utilized and the water is conducted through a long head race, if land of a suitable elevation is available, or, otherwise, through a tunnel or elevated flume to a point immediately above the site of the power station. From the end of the tail-race or tunnel or flume the water is carried to the plant through a pipe line.

Fourth: The fourth type is similar to the third except that where the head-race or tunnel is used (the ground being unfavorable to such construction or the expense of the same being unwarranted) a long pipe line is provided to conduct the water from the head works to the station.

Fifth: The fifth type is the tunnel tail-race type and involves conducting the water through pipe lines directly to the wheels, from which water is discharged into a tunnel tail-race through which the water is discharged back into the stream.

It is important to note in this case, as in all other cases of attempted classification, that such classification is for the purpose of a systematic grouping of numerous diversified types to facilitate study and investigation. In the actual adaptation of plans of de-

velopment, single types are seldom found and modifications of types are essential.

286. Concentrated Fall.—In most of the low head water powers the portion of the fall of the river which can be utilized is distributed over minor rapids and small falls and occupies a considerable length of the stream. Where the head is small and the expense of a dam to concentrate the head entirely at one point is permissible, the power house may perhaps be located to advantage in the dam. In this case the power house will constitute a part of the dam. This is possible only where the length of the spillway remaining is sufficient to pass maximum flood without an undue rise in the head of the water above the dam. In many such cases this plan, which is represented by Diagram C, Fig. 373, page 591, meets economical construction, as it may both cheapen the cost of the dam and reduce the excavation necessary for the wheel pit and raceways. The power house built at such point is, however, usually directly in the line of the current and must be so constructed and protected as to prevent its injury or destruction by floods, ice or other contingencies of river flow.

In other cases, where the spillway available by the above plan is not sufficient, or where the plant is not properly protected by such a plan of construction, the plant may be constructed on one side of the dam, receiving its waters from a head-race which joins the river above the dam and discharges it into the river below, as shown by Diagrams C and D, Fig. 373. Or, where the conditions are suitable, the plant may receive the water through a head gate from the river above the dam and discharge it through a tail-race which may enter the river at some point below the dam, as shown in Diagram A, Fig. 373.

In cases where the power is to be distributed to a number of independent plants, raceways may be constructed on either or both sides of the stream and from the dam, following the stream downward along the bank and more or less approximately parallel thereto as the nature of the conditions demand. The plant drawing the water from this head-race may be distributed at various points along the same, and from these plants the water will be discharged after use either directly into the stream itself or into a tail-race connecting such plants with a lower point farther down the stream, as shown in Diagram E, Fig. 373.

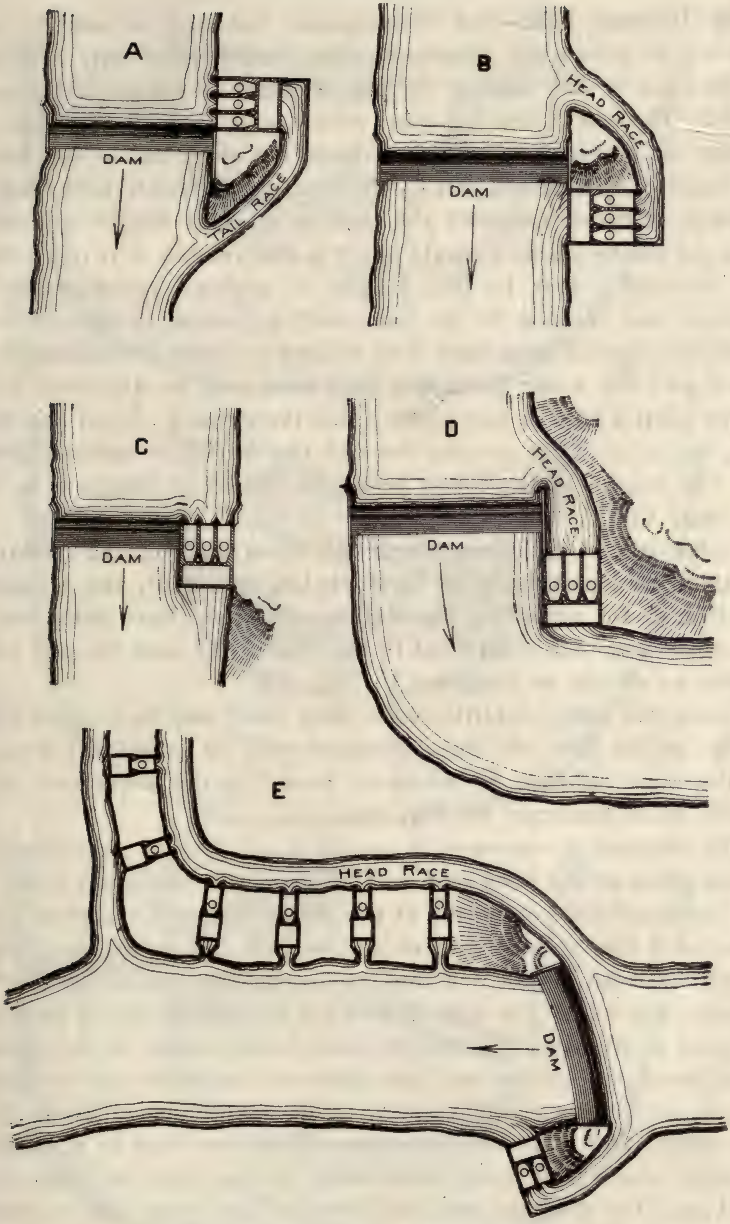

Fig. 373.—Typical Arrangements of Hydraulic Power Developments.

287. Divided Fall.—An independent tail-race is usually constructed to advantage where the dam concentrates only a portion of the head or fall, leaving certain additional portions to be developed by the use of the tail-race, which may, if desirable, enter the stream at a point much farther down the river and at the foot of the rapids. Where the fall of the stream is considerable, and the expense of construction of the dam to suitable height to concentrate the entire fall at a single point is inadvisable, it is often desirable to build a dam to less height at perhaps considerably less expense and develop at the dam only a portion of the total fall. From this dam a head-race may extend to some considerable distance, and the water from this head-race may be delivered to the power plant a mile or two lower down the stream. From this head-race, the water, after passing through the wheels, is carried directly into the stream at the lower point, as shown in Diagram G, Fig. 374, page 593.

Under other conditions, where the topography of the country is suitable, the head-race may be much less in extent, and a tail-race substituted for receiving the waters after they have been used in the wheel and then conducted to the river at or near the end of the rapids, as shown in Diagram F, Fig. 374.

Under still other conditions the plant itself may be located immediately at the dam and the tail-waters may be conducted from the turbine to a tail-race or tail-water tunnel to the lower end of the rapids, as in Diagram H, Fig. 374.

The relation of raceways is merely a question of developing the power plant at the least cost and securing the maximum head, and the topographical conditions at the power site will therefore determine what line of development will be best. In a number of cases, where the head or fall is considerable and the power development is large, and where the cost of land for head-races would be almost or quite prohibitive, the stations have been located in the immediate vicinity of the river and have delivered the water into a tail-race tunnel, which frequently empties at a considerable distance down the stream and at the lowest point of delivery that is practicable. In other cases it is more economical to run open raceways for a portion of the distance and then conduct the water under pressure by closed pipes to the wheels at the lower point.

This last method is used particularly under high head and where the water must be conducted for a reasonable distance over an irregular profile.

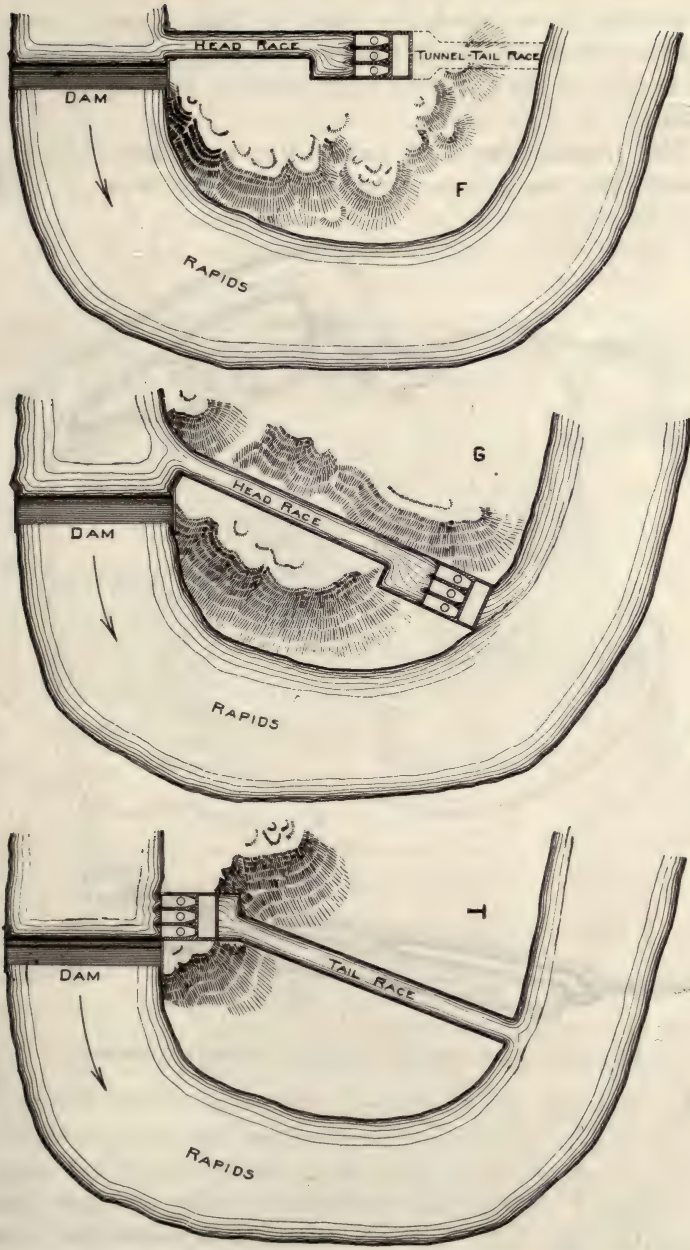

Fig. 374.—Typical Arrangements of Hydraulic Power Developments (see page 592).

The quantity of water to be used, the head available, and the value of power modify the arrangements which must be carefully studied in view of the financial, topographical, and other modifying conditions.

288. The Distribution of Water at Various Plants.—Figure 375 is a plan of the power development on the Rock River at Sterling,

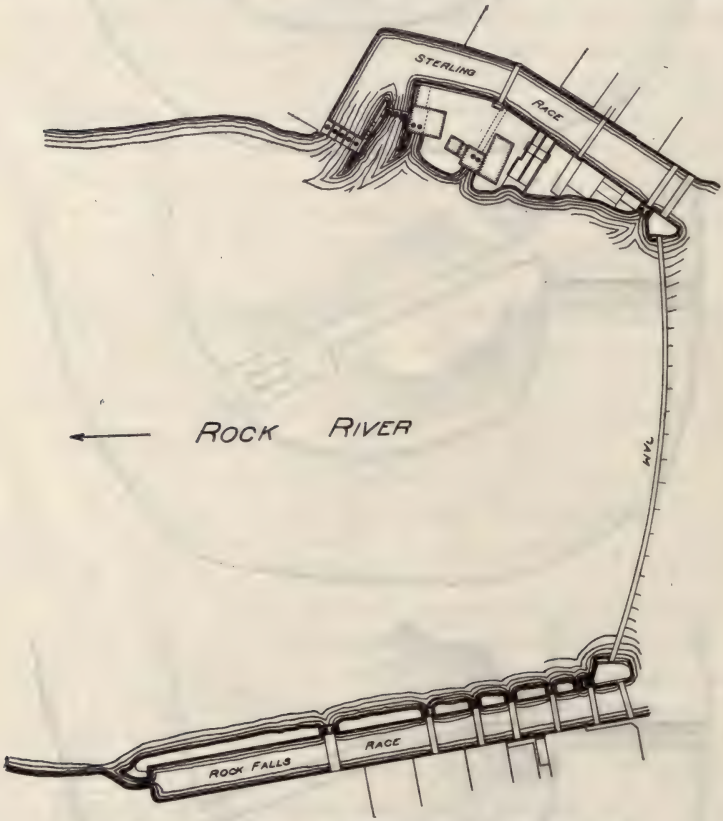

Fig. 375.—Raceways of Sterling Hydraulic Company.

Illinois. The dam at this point is about 940 feet in length. The power is owned by various corporations and private individuals who have combined their interests in the dam and raceways and have organized the Sterling Hydraulic Company, whose function is to maintain the same. The individual plants are owned, installed, and operated by the various owners or by manufacturers who lease the power. At this location races have been constructed at the foot

of the rapids, but these rapids continue to a point near the lower end of the tail-race, and the plants farthest from the dam have the highest falls. The fall varies from about eight to nine and one-half feet.

Figure 376 shows the general arrangement of the canal of The Holyoke Water Power Company at Holyoke, Mass. The total fall of the river at this point, from the head water above the dam to the tail-water at the lowest point down the stream, is about sixty feet. The fall is divided into three levels by the various canals, marked: first level canal, second level canal, and third level canal.

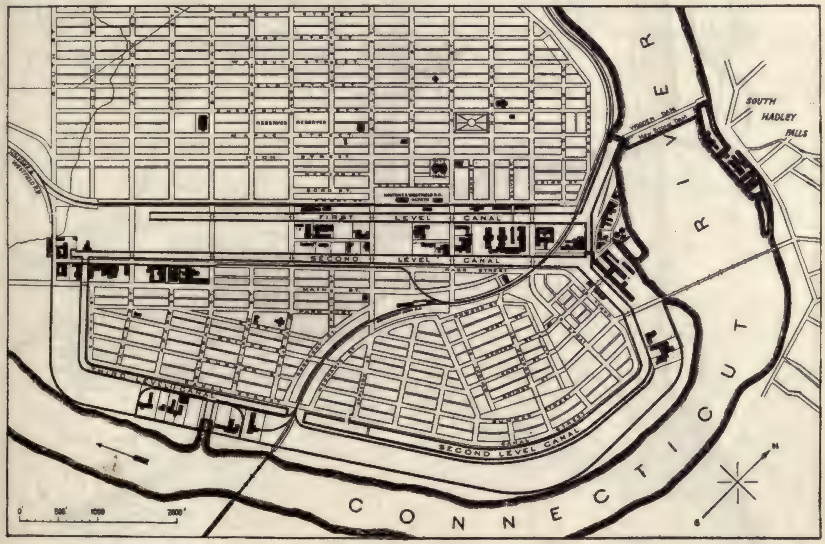

Fig. 376.—Canals of Holyoke Water Power Company.

The first level canal, which has a length of about 6,000 feet, is constructed as a chord across the bend of the river and is approximately some 3,000 feet from the bend. The canal is about 150 feet wide near the bulkhead and decreases to about 100 feet at the lower end. The water depth is about twenty feet at the upper end and about ten feet at the lower. The canals are all walled throughout their length to a height two or three feet above the maximum water surface. The fall from the first level to the second is about twenty feet. Various mills draw their water supply from the first level as a head-race, and discharge into the second canal as a tail-race. Near the upper end of the canal are a few factories that draw water

from the first level and discharge the same into the river with a head of some thirty-five or forty feet.

The second level canal is built parallel to the first and at a distance of about 400 feet nearer the river. The main canal is about 6,500 feet in length, but near the left hand of the map is shown to sweep round towards the river and attain a reach of about 3,000

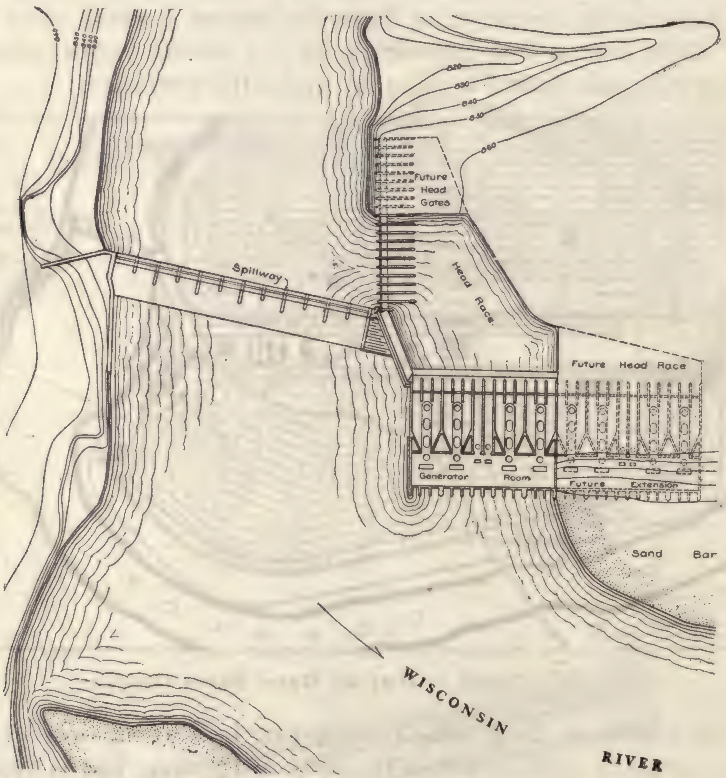

Fig. 377.—Kilbourn Plant of Southern Wisconsin Power Co. (see page 597).

feet in length parallel thereto. The mills drawing their supply from this canal discharge either directly into the third level or into the river. The water supply from each of the lower levels is the tail-water from the next level above, but is also supplemented by overflows when the mills fed from the level above are not discharging sufficient water to maintain the quantity needed in the lower level.

The fall from the third level of the river is essentially the same for all the mills drawing water therefrom, but according to the stage of the river ranges from fifteen to twenty-seven feet.

The flow of water in the first level is controlled by gates and its height limited by an overflow of about 200 feet in length, which acts as a safety overflow and prevents any great rise in the head water during times of flood.

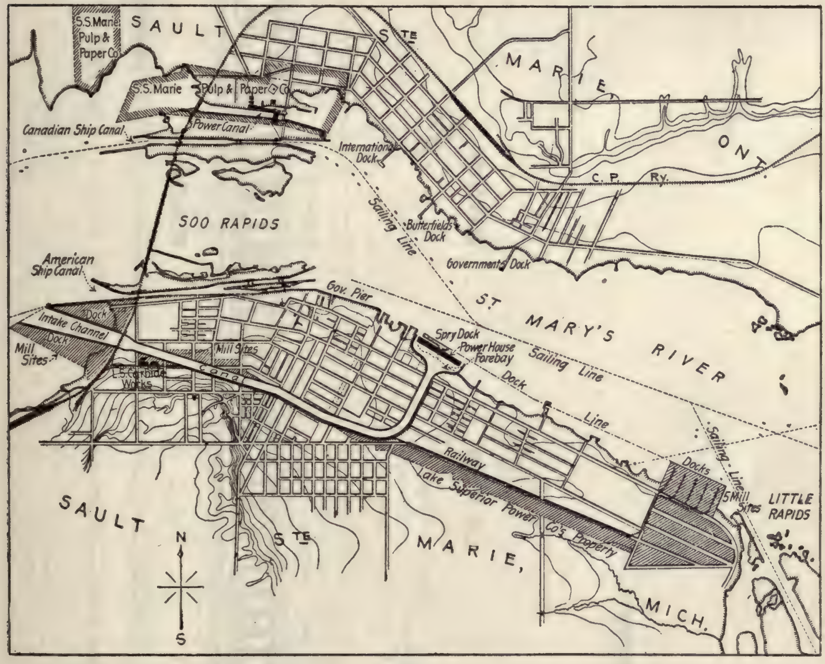

Fig. 378.—Plant of The Lake Superior Power Co. (see page 601).

289. Head-Races Only.—Fig. 377, page 596, illustrates the general plan of the hydraulic power development of The Southern Wisconsin Power Company at Kilbourn, Wisconsin. Here the entire cross-section of the stream is necessary in order to pass the maximum volume of water, which amounts to about 60,000 second-feet. The plant has therefore been constructed at one side of the river, receives the flow through a series of gates built just above the dam, and discharges the water into the river just below the bend in the river (see Fig. 377 and Frontispiece). The plant now constructed is only a portion of that which it is designed to ultimately

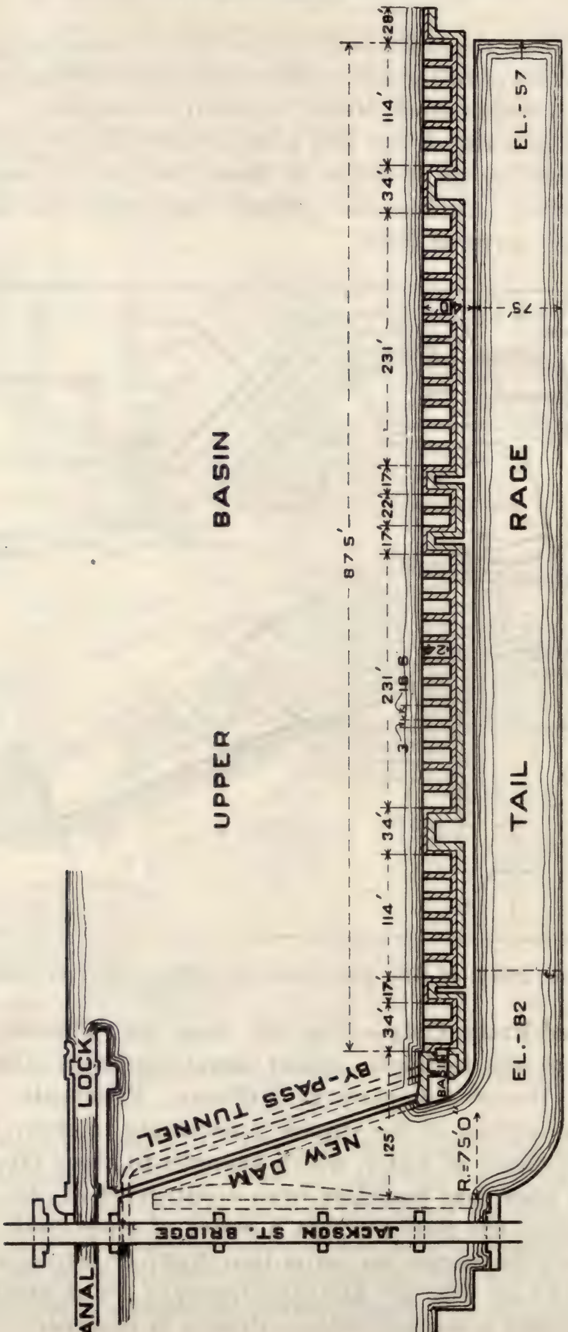

Fig. 379.—Joliet Plant of Economy Light and Power Company (see page 601).

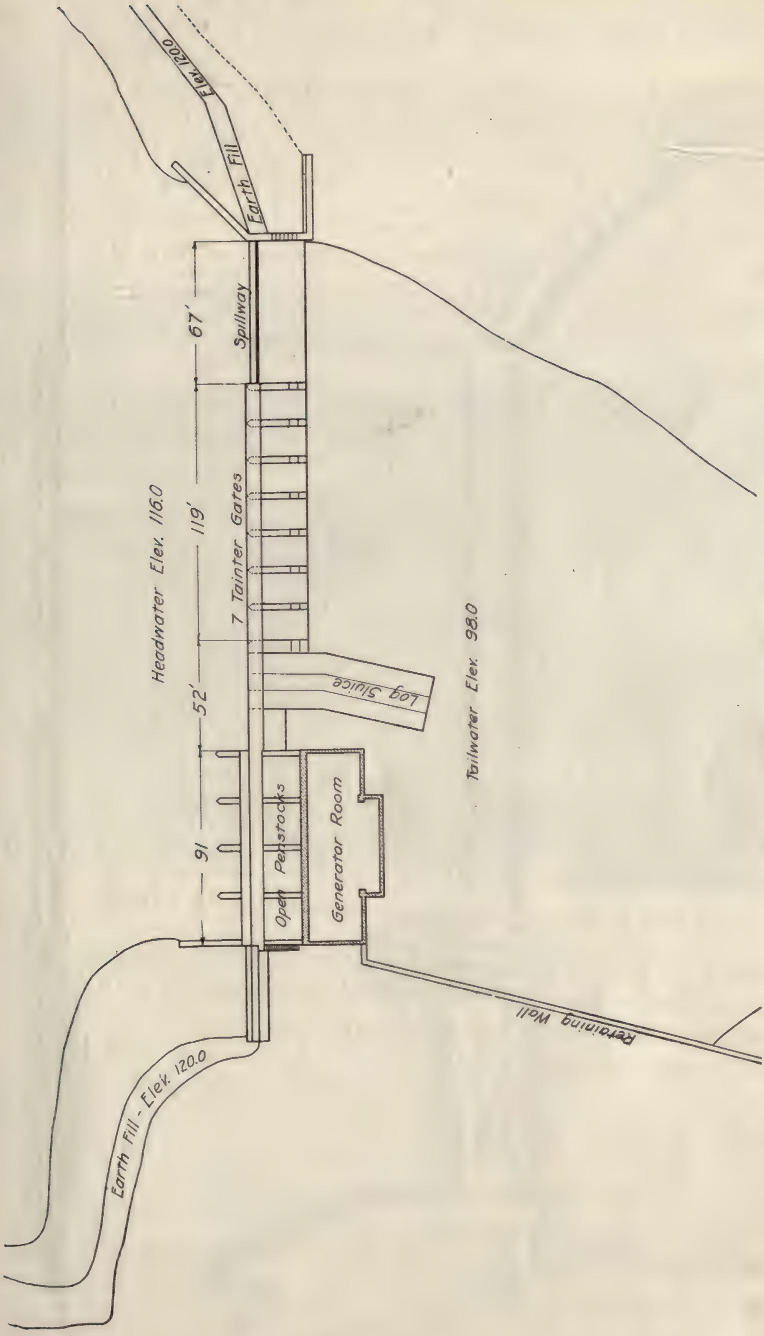

Fig. 380.—Development of the Alpena Power Co. at Alpena, Mich. (see page 603).

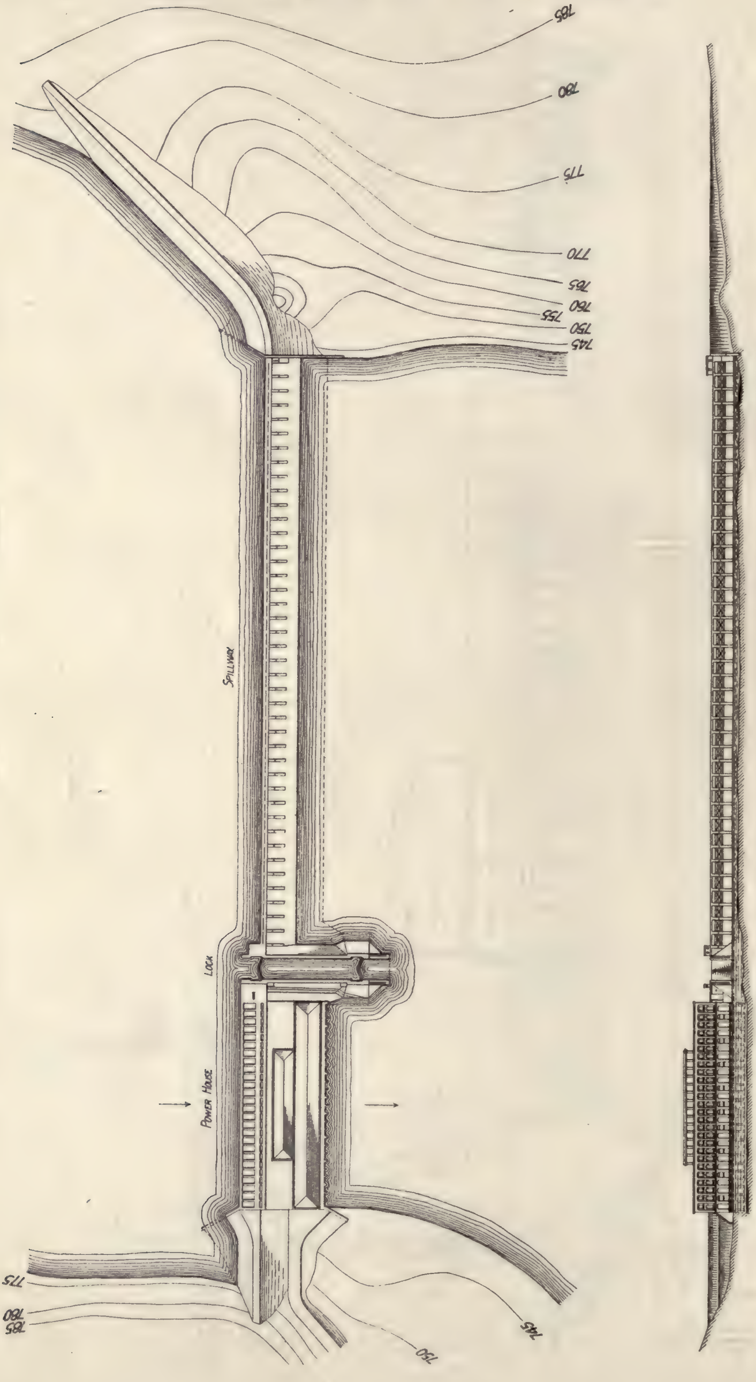

Fig. 381.—Development of the Wisconsin River Power Co. at Prairie du Sac, Wis. (see page 603).

install. The proposed future extension of the power plant is shown by the dotted lines.

Figure 378, page 597, shows the water power plant of The Lake Superior Power Company at St. Mary's Falls, Michigan. The canal on the American side begins just above the entrance to the American ship canal and above the Soo rapids. The water is conducted through this canal to a power house located below the rapids at the point shown on the map. On account of the value of the

Fig. 382.—Prairie du Sac Plant of the Wisconsin River Power Co.

land this canal was designed for a velocity of flow of about seven and one-half feet per second with full load of the plant, which was designed for about 40,000 H. P. requiring a capacity with available head of 16.2 feet, of about 4,200 cubic feet per second. (See Engineering News of August 4th, 1898.)

Figure 379, page 598, shows the plan of the hydraulic development of The Economy Light and Power Company at Joliet, Illinois. The entire installation as shown is owned by this company. The fall available is about eleven feet and is developed by a concrete dam which creates the upper basin along which the power plant has been constructed. The water flows through the flume

Fig. 383.—Possible Canal for Peshtigo River Development (see page 604).

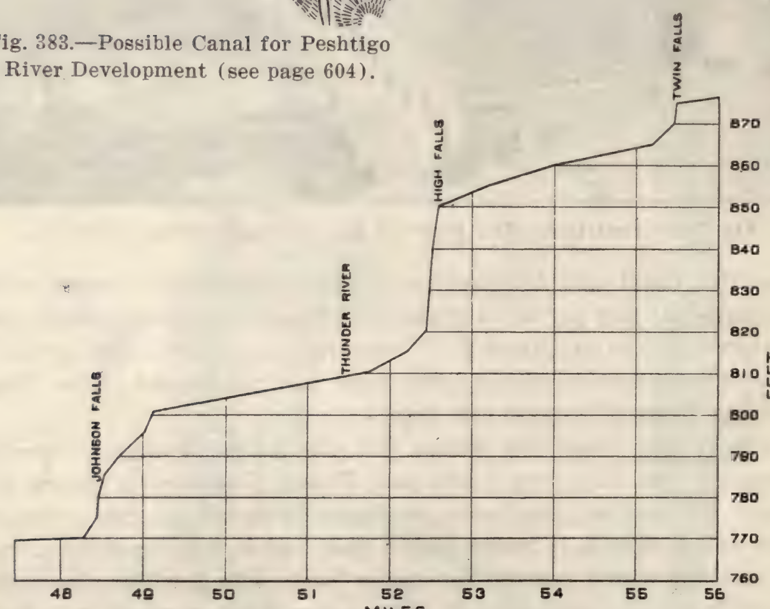

Fig. 384.—Profile of Peshtigo River (see page 604).

gates directly on to the wheels and is discharged into a tail-race built parallel with the river. A certain amount of water is necessary for feeding the lower level of the canal and this is supplied by a by-pass tunnel shown in dotted line above the dam. This by-pass, which is slightly higher than the elevation of the rail-race, is fed by the discharge of one of the wheels, which operates under a less head than the other wheels in the installation.

290. Plant Located in Dam.—Fig. 380, page 599, is a plan of the spillway, tainter gate section, log sluice and power house that constitute the dam of the Alpena Power Company on the Thunder Bay River at Alpena, Michigan.

Fig. 385.—Plant of the Wisconsin Public Service Co. at High Falls, Wis. (see page 604).

In Fig. 381, page 600, is shown the general plan and elevation of the hydraulic plant of the Wisconsin River Power Company at Prairie du Sac, Wisconsin.

In this case the dam is built across a wide channel, and the total breadth of the river is much greater than necessary to accommodate the flood flow of the stream which is approximately 75,000 second-feet. In consequence, the power plant forms a part of the dam itself and the spillway will occupy only a portion of the entire length of the structure. The head of the water above the dam is controlled by forty-one tainter gates, by means of which the level of the water above the dam can be controlled at all stages of flow (see Fig. 359, page 566, and Fig. 382, page 601).

291. High Head Developments.—Fig. 383, page 602, illustrates the general plan of a possible method of development of the Peshtigo River Company. The fall available is shown by the profile given in Fig. 384, page 602.

The plan originally contemplated the construction of a dam above High Falls of sufficient height to back the water over Twin Falls, and to either develop the power at High Falls and Johnson's Falls independently or conduct the water by a canal to Mud Lake, thence to Perch Lake, thence to the head works to be built above Johnson's Falls, where a head of about 110 feet would have been available. Legal complications in regard to the diversions of water prevented the single development, and a plant utilizing about 80 feet of head has been constructed at High Falls.

This plant is located directly in the old stream bed at the foot of the falls, receives its water directly from the reservoir through several steel penstocks, and discharges it through draft tubes directly into the river below. The spillway and six tainter gates, which control the reservoir level during high water, discharge into a new channel east of the former falls (see Fig. 385, page 603).

Figure 386 shows the plan of the power development at Trenton Falls, New York. The upper portion of the fall is developed by a dam about sixty feet in height, which is connected by an eighty-four inch pipe line with the turbine located in the power house about two miles below. The turbines used in this development are the Fourneyron

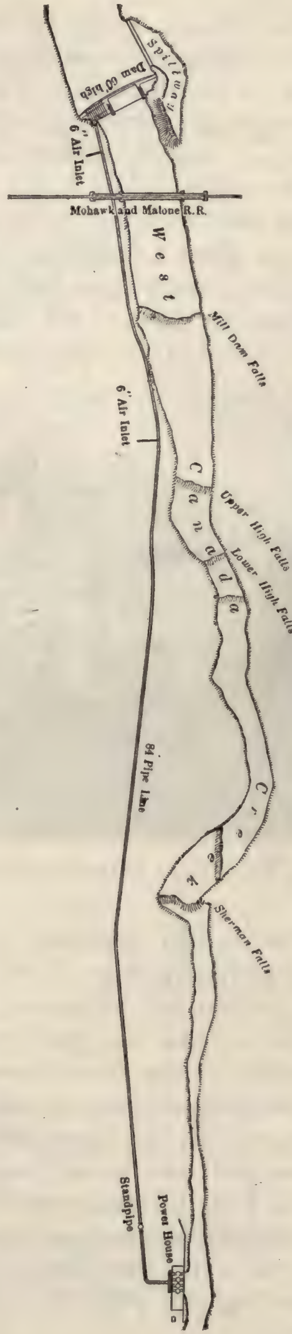

Fig. 386.—Plan of Power Development at Trenton Falls, New York.

turbines, which are described in Chapter XV, and are illustrated by Fig. 322, page 514.

Figure 387 is a general plan of the water power developments at Niagara Falls. The first development was that of The Niagara Falls Hydraulic and Manufacturing Company. By means of a canal the water is taken from the upper end of the rapids and

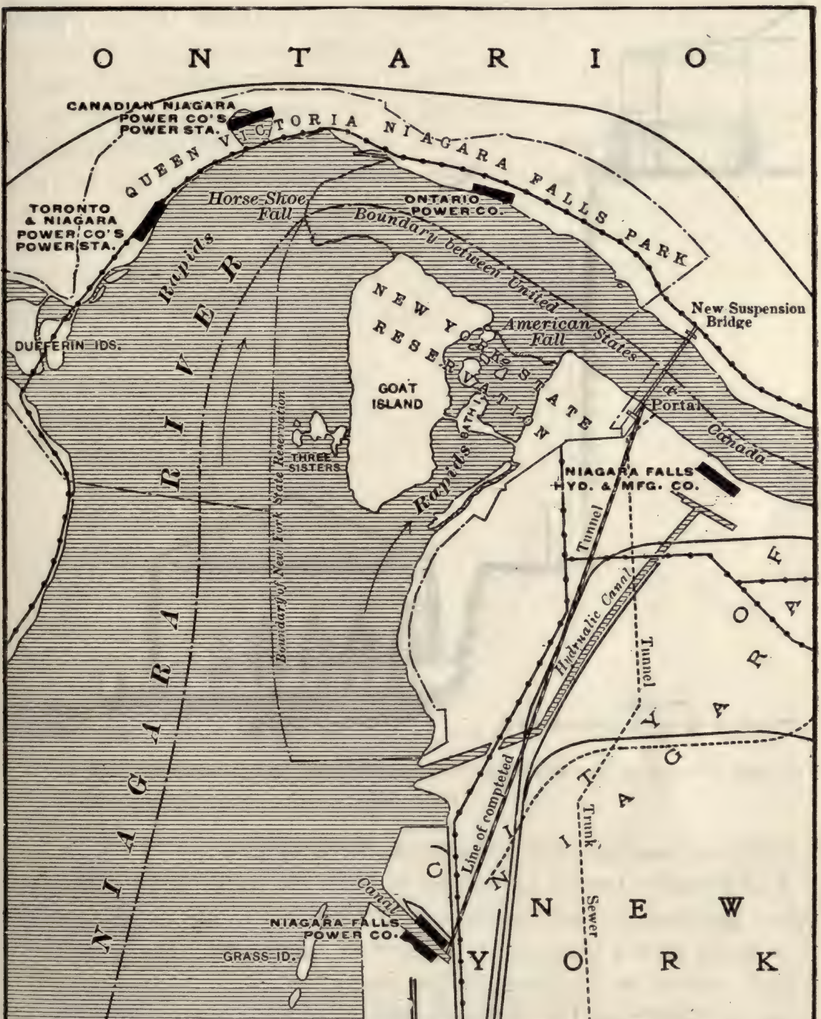

Fig. 387.—Niagara Falls Power Development.

conducted to the lower bluff on the American side, and distributed, by open canals, to various plants located along this bluff.

The second plant constructed was that of The Niagara Falls Power Company, in which power is developed by the vertical shafts connecting with a tail-water tunnel which discharges into the river just below the new suspension bridge.

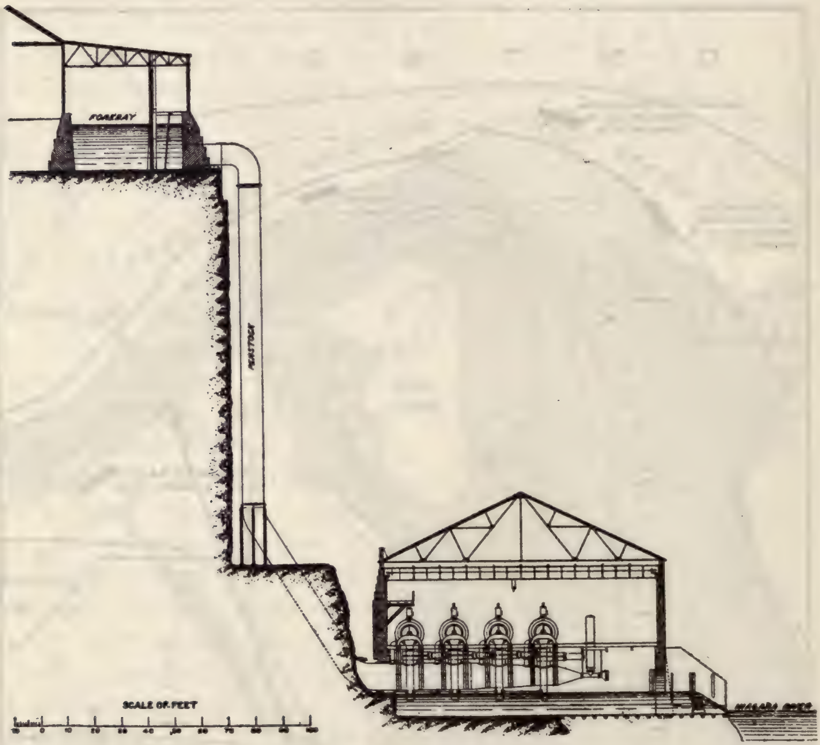

Fig. 388.—Plant of the Niagara Falls Hydraulic and Manufacturing Co. (see page 607).

On the Canadian side are shown three plants.

The Ontario Power Company secures its water supply from the upper portion of the rapids, conducting it through steel conduits to a point above the power house and thence by penstocks to the wheel, located in the gorge below the falls.

In the plants of The Toronto and Niagara Power Company and The Canadian-Niagara Power Company, the water is taken from above the Falls and discharges through penstocks to wheels located

at the base of a shaft and thence into tunnels, discharging into the river at a point below the Falls.

Figure 388, page 606, illustrates the plant of The Niagara Falls Hydraulic and Manufacturing Company, which is supplied by water from the hydraulic canal above mentioned. The water is conducted from the forebay by a vertical penstock to which is attached several wheels which deliver the water into a tail-race tunnel and thence into the gorge below.

The plant arrangments above described are typical of many now in use both in this country and in Europe. It is at once obvious that in considering this subject each particular location is a problem by itself which must be considered in all its bearings; but an understanding of the designs and arrangements already in use forms a satisfactory basis from which a judicious selection can be made with suitable modifications to take care of all the conditions of topography and other controlling conditions.

CHAPTER XX

PRINCIPLES OF CONSTRUCTION OF DAMS

292. **Object of Construction.**—A dam is a structure designed to hold back or obstruct the flow and elevate the surface of water. Such structures may be built for the following purposes:

First: To concentrate the fall of a stream so as to admit of the economical development of power.

Second: To deepen the water of a stream so as to facilitate navigation and to so concentrate the fall that vessels may be safely raised from a lower to an upper level by means of locks.

Third: To impound or store water so that it may be utilized as desired for water supply, water power, navigation, irrigation, or other uses.

Fourth: In the form of mine dams or bulk heads to hold back the flow of water which would otherwise flood mines or shafts or cause excessive expense for its removal.

Fifth: As coffer-dams for the purpose of making accessible, usually for construction purposes, submerged areas otherwise inaccessible.

293. **Dams for Water Power Purposes.**—The primary object of a dam constructed for water power purposes is to concentrate the fall of the stream so that it can be developed to advantage at one point and so that the water thus raised can more readily be delivered to the turbines through raceways and penstocks of reasonable length. This object is occasionally accomplished for small developments in rivers with steep slopes and high velocities by the construction of wing dams which occupy only a portion of the cross-section of the stream, but cause a heading up of the water and direct a certain portion of the flow into the channel or raceway through which it flows to the wheels. In streams of moderate slope the dam must extend entirely across the stream in order to concentrate sufficient head to be of practical utility.

Wing dams can be used at the head of high falls where only a portion of the volume of flow can be utilized, as at Niagara Falls, or in rapid rivers where a portion of the flow is to be directed into a narrow channel for utilizing low heads by means of undershot or

float wheels, as is frequently done for irrigation purposes. Where the full benefit of both head and volume is to be utilized the dam must extend from bank to bank and be constructed to as great a height as possible.

294. Height of Dam.—To utilize a river to the maximum extent the highest dam practicable must be constructed.

The height of a dam may be limited by the following factors:

First: The overflow of valuable lands above the dam site.

Second: The interference with water power rights above the point of development.

Third: The interference with other vested or public rights.

Fourth: The cost of the structure.

The value of the power that can be developed by means of a proposed dam will limit the amount that can be expended in the purchase or condemnation of property affected by backwater from the dam and the cost of its construction. These are among the elements of the cost of the project and must be considered together with other financial elements before a water power project can be considered practicable.

In considering backwater and its effect on riparian rights both high and moderate conditions of flow must be considered. The former condition gives rise to temporary interference, often of little importance when affecting purely farming property, and the real or fancied damages can be liquidated by securing releases from the owners. The latter condition will permanently inundate certain low lands which must be secured by purchase or condemnation. In many states where the laws of eminent domain do not apply to the condemnation of property for such purposes it is necessary to secure such property by private purchase before the work is undertaken, and usually before the project becomes known publicly, for in such cases the owner of a single piece of land may delay the project by a demand for exorbitant remuneration, from which demand there is no escape in such cases. In every case it is desirable that riparian and property rights be fully covered before the construction of the works actually begins.

295. Available Head.—Beside the question of backwater the question of head at the dam is important both in relation to the question of interference and in relation to the question of power. In relation to interference it is an easy matter with a known length and height

of dam to determine by calculation from a properly selected weir formula the height of water above the dam under any condition of flow. To determine the head available under all conditions of flow the weir curve must be studied in connection with the rating curve as discussed in Chapter VIII.

Two conditions of flow often require consideration in this connection:

First: Where a considerable portion of the flow is being utilized by the wheels and therefore does not affect the head of the dam.

Second: Where the water is not being used by the wheels and consequently affects the head of the dam.

Both of these conditions should be studied and determined in relation to their influence on both backwater conditions and power.

296. The Principles of Construction of Dams.—The general principles for the construction of all dams are similar, and are as follows:

First: They must have suitable foundations to sustain the pressure transmitted through them, which must be either impervious or rendered practically so.

Second: They must be stable against overturning.

Third: They must be safe against sliding.

Fourth: They must have a sufficient strength to withstand the strains and shocks to which they are subjected.

Fifth: They must be practically water-tight.

Sixth: They must have essentially water-tight connections with their bed and banks, and, if bed or banks are pervious, with some impervious stratum below the bed and within the banks of the stream.

Seventh: They must be so constructed as to prevent injurious scouring of the bed and banks below them.

The application of the above principles depends on the material from which the dam is to be built and on local conditions.

297. The Foundation of Dams.—The materials used for the construction of dams may be masonry, which includes stone-work and concrete-work, reinforced concrete, timber, steel, loose rock, and earth. Each may be used independently or in combination. Masonry and concrete dams must be built upon foundations which are practically free from settlement. Masonry dams for heads of perhaps forty feet may be safely constructed on proper pile founda-

tion or sands and gravels, with proper cut-off precautions, but materially higher dams can be safely built only upon solid rock. Reinforced concrete is now being extensively used for small structures and is not as seriously affected by slight settlement as in the case of dams of solid masonry. There is, however, little flexibility in structures of this kind, and the foundation must be selected in accordance with this fact. Timber and steel possess a flexibility not possible in concrete construction and are much better adapted to locations where the foundation may be subject to settlement.

In construction on rock foundation it is usually desirable to excavate trenches therein in order to give a bond between the structure of the dam and its foundation. It is also essential with rock foundations to determine whether cracks or fissures in the foundation extend below the structure, and if such are found, they must be completely cut off.

On earth, sand or gravel foundations, when such must be used, the flow which would take place through these materials and under the structure of the dam must be completely cut off by the use of steel or timber sheet piling, which, if possible, should be driven from the structure to the rock or to some other impervious stratum. If no impervious stratum is accessible, the sheet piling must be driven to such a distance below the base of the dam that the friction of the flow of water under it will reduce or destroy the head and consequently reduce the flow of water to an inappreciable quantity (see Fig. 402, page 625).

298. Strength of Dams.—A dam to be built in a flowing stream should be designed with a full appreciation of all the stresses to which it may be subjected. Of these, stresses that are due to static pressure can be readily estimated from the known conditions. The strains due to dynamic forces are not so fully understood or easily calculated. Where the structure is constructed to retain a definite head of water without overflow, as in the case of reservoir embankments, the problem becomes one largely of statics and the only other stresses to be considered are those due to ice action and the action of waves on the structure. When a dam is constructed in a running stream and is subject to the passage of extensive floods of water over it, frequently accompanied by large masses of floating ice, logs or other material which in many cases may strike the crest of the dam, and bring unknown and violent strains, the prob-

lem becomes largely one of experience and judgment.

299. Flood Flows.—Every dam must be provided either with a spillway of sufficient capacity to pass the maximum flood or with waste gates by means of which such floods can be controlled. A considerable height of water is necessary in order to pass a large volume of water over a limited spillway, and such construction usually causes too great a reduction of head during the low water period. Large capacity is often best secured by the use of a series of gates which may occupy a part or the entire section of the dam (see Frontispiece and Fig. 381, page 600). Perhaps there has been no more frequent cause for the failure of dams than inadequate spillways. Figure 389 shows the dam at Black River Falls, Wisconsin, during the June flood of 1911, when a discharge of about 10,000 cubic feet per second more than filled the spillway provided. In October of the same year, a flood of perhaps 40,000 cubic feet per second augmented to about twice that amount by the failure of two large reservoirs on the Upper Wisconsin River, cut around the north end of this dam and destroyed most of the business district of

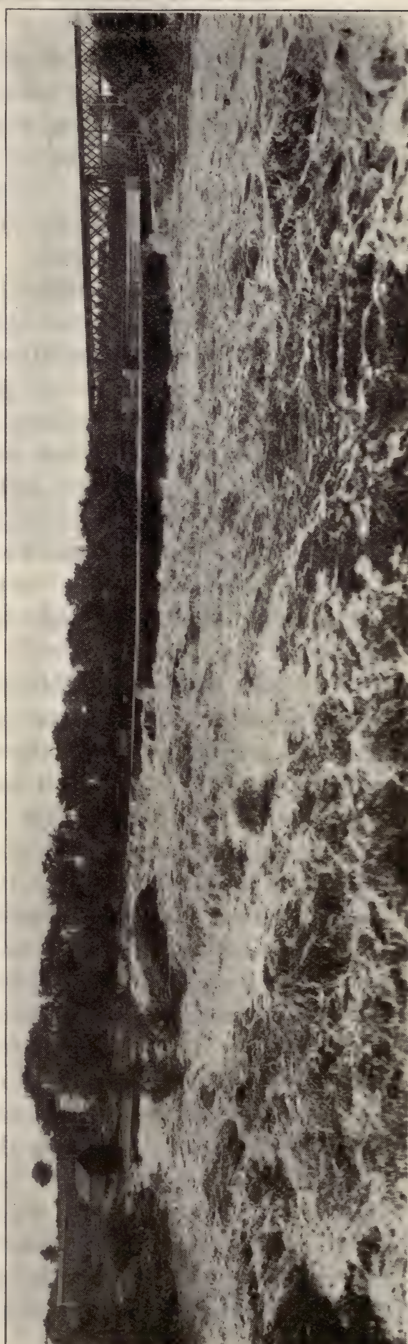

Fig. 389.—Dam at Black River Falls, Wis., During Flood of June, 1911.

Black River Falls, with a loss of $1,000,000 or more. This dam was later replaced by the gate section shown in Fig. 390.

The passage of great volumes of water over a dam involves the expenditure of the power so generated upon or immediately adjoining the structure, and unless preparations are made for properly taking care of this immense expenditure of power, the power may be exerted in the destruction of the structure itself.

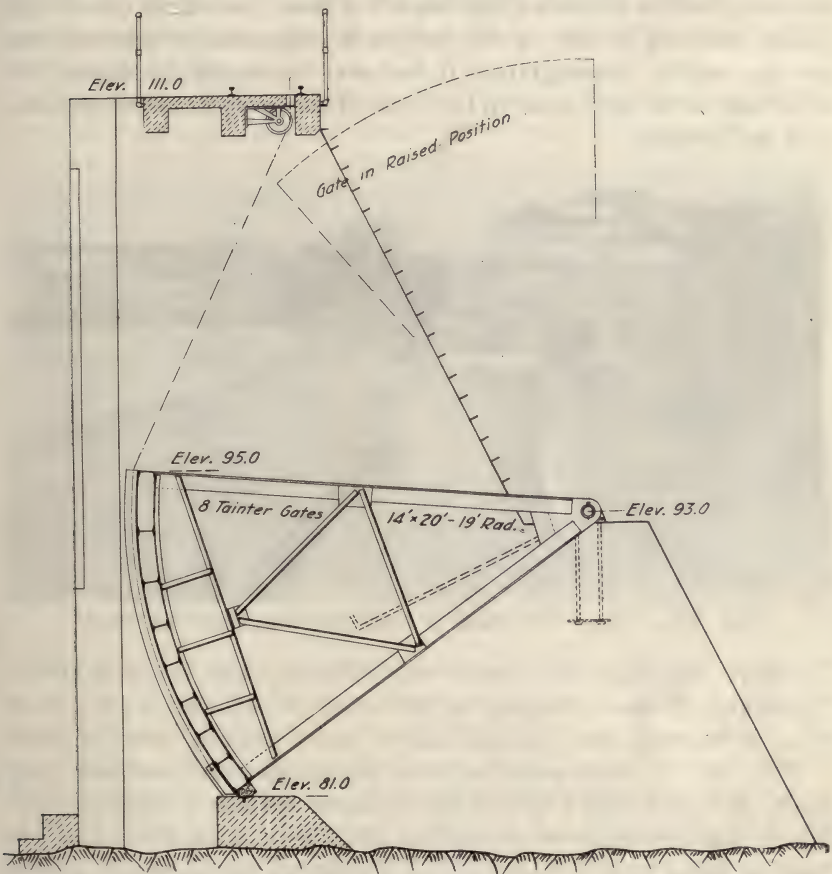

Fig. 390.—Tainter Gate Section of the New Dam at Black River Falls.

Figures 391 to 393 show three views of the timber crib dam at Janesville, Wisconsin, under various conditions of flow. In Fig. 391, page 614, the flow of the river is comparatively small and all of the water is being used in the power plant, none passing over the dam. In Fig. 392, page 614, the river is at a moderate stage and

the greater part of the flow is passing over the crest of the dam. In Fig. 393, page 616, some four or five feet of water is passing over the dam and the power that is developed thereby is causing the standing wave and the rough water shown in the picture below the dam. At this point the power developed by the fall is being expended in waves and eddies, which, unless properly controlled, will attack and injure or destroy the structure. On rock bottom the rock itself will sustain the impact of flow over small dams. But where the rock is soft, or the bottom is composed of material that can be readily disintegrated, it becomes necessary to extend the structure of the dam itself in the form of an apron to cover and protect the bottom.

Fig. 391.—Timber Crib Dam at Janesville, Wis. (see page 613).

Figure 394, page 617, shows the Kilbourn dam of the Southern Wisconsin Power Company at Kilbourn, Wisconsin, with a flood flow of fourteen and one-half feet of water passing over its crest.

This dam is constructed with an adjustable crest and was originally provided with a timber apron about 100 feet in length. This apron was gradually destroyed by the heavy flood flows and has been replaced by a shorter but heavy concrete apron, shown in Fig. 411, page 651.

The two ends of the Kilbourn dam rest directly upon rock foundation. The center of the dam is supported by cribs constructed to fit the rock bed.

At the upper face of the dam triple sheeting were driven, closely fitting the rock bed, and securely fastened to the dam and cribs

from the rock up, thus practically preventing any considerable flow of water through the dam.

During high floods the amount of power expended in the passage of water over the dam will exceed 60,000 horse power. The entire surface of the dam exposed to air at times of low water is constructed of reinforced concrete, attached directly to the timber work by steel reinforcement. By this design a structure is obtained having all the advantages of the flexibility of timber, with the lasting qualities of masonry, all timber work being submerged under every ordinary condition.

Figure 395, page 618, shows the dam of the Marathon Paper Mills at Rothschild on the Wisconsin River. This dam was built

Fig. 392.—Janesville Dam With Moderate Water (see page 613).

of concrete on a pile foundation, and was constructed with an apron to protect the sand and gravel deposits below it, built of concrete slabs anchored together by I-bar joints built into the slabs. No sheeting was placed below the apron, and it was believed that if underwashing occurred, the slabs would settle and limit the extent of the undermining. The heavy flow of flood water through the tainter gate section destroyed the apron, as shown in Fig. 395. The apron has since been replaced by one of heavy timber construction.

300. Impervious Construction.—Masonry dams are commonly made impervious by the structure of the masonry itself.

In timber crib dams ordinarily no attempt is made to make the structure itself water-tight, but the top and upstream side are usu-

ally covered with water-tight sheeting to prevent the water passing into and through the cribs. Such water as reaches the timber cribs usually passes away readily through the open structure on the down stream side of the dam.

In the construction of rock filled dams the same condition ordinarily obtains. The dam is fairly porous, with the exception of its upper face which is made practically water-tight by the use of concrete, puddle, or some impervious paving.

In earthen dams the finer and more water-tight materials are used on the inner slopes of the embankment, and, in addition thereto, it is customary in large and important works to use a core

Fig. 393.—Janesville Dam Under High Water (see page 613).

of concrete or puddle to effectively prevent the passage of water through the structure.

Pervious foundations may be made water tight by the use of a sheet piling driven well below the river bed, and by the extension of the base of the structure so that the head will be destroyed by the necessary length of flow of seepage water. The foundations of the dam of the Olympian Power Company on the Elwha River, was successfully replaced, after the lower section was washed out under a 100 foot head, by a rock filled base work 300 feet long, with five lines of thirty foot steel sheet piling below the fill and the upstream face made practically impervious by small rock, gravel and clay. (See Engineering Record, March 28, 1914.)

301. The Stability of Masonry Dams.—The external forces acting on a masonry dam are the water pressure, the weight of the

masonry, the reaction of the foundation, ice and wave pressure near the top, wind pressure, and back pressure of the water on the down stream side. The action of these forces may cause a dam to fail by:

(1) Sliding on the base or on any horizontal plane above the base.

(2) Overturning.

(3) Crushing the masonry or foundation.

If the dam be built of rubble masonry there will be no danger of failure by sliding on a horizontal joint above the foundation and experience has shown that where a good quality of mortar is used it can be depended upon to prevent sliding in concrete and stone

Fig. 394.—Dam of the Southern Wisconsin Power Co. at Kilbourn, Wis.

dams having horizontal bed joints. The joint between the dam and its foundation is a more critical point. In rock foundation steps or trenches should be cut so as to afford good anchorage for the dam. In the case of clay, timber or similar foundations the dam will have to be made massive enough so that the tangent of the angle between the resultant pressure on the base and a vertical line is less than the co-efficient of friction between the materials of the dam and the foundation.

It is customary in the design of masonry dams to proportion the section so that the lines of resultant pressure at all horizontal joints,

for both the conditions of reservoir full and reservoir empty, shall pass through the middle third points of the joints. If this condition is fulfilled, the factor of safety against overturning at every joint will be two, and there will also be no danger from tensile stresses developing in the faces of the dam.

Investigation has shown that there is no danger of crushing the masonry except in very high dams, with the consideration of which we are not here concerned.

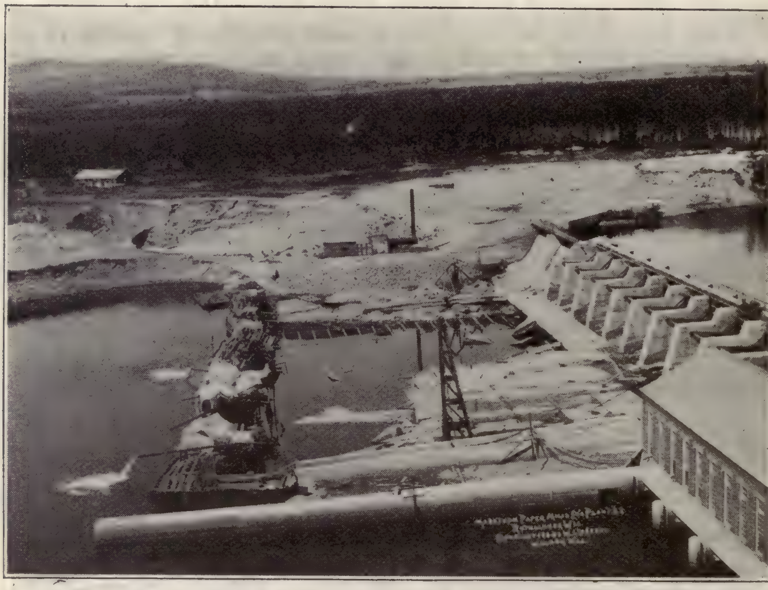

Fig. 395.—Dam of the Marathon Paper Mill on the Wisconsin River at Rothschild, Wis. (see page 615).

302. Calculation for Stability.—The general conclusion may therefore be stated, that, in the case of ordinary masonry and concrete dams, not over 100 feet in height, to be built on rock foundations, the design can be based upon the condition that the lines of pressure must lie within the middle third of the profile. This rule must be modified at the top of the dam to resist the stresses due to waves, ice, etc. The force exerted by ice is an indeterminate quantity and the tops of dams must therefore be proportioned in accordance with empirical rules. Dams are built with top widths varying from two to twenty-two feet, the broader ones usually carrying a roadway.

Coventry suggests the following empirical rules for width of top above water level:

$$b = 4.0 + 0.07\,H \qquad (238)$$

$$y_0 = 1.8 + 0.05\,H \qquad (239)$$

Where b is the width of top, y_0 the height above water level and H the greatest depth of water, both faces of the dam will be vertical until the depth y_1, is reached, where the resultant force passes through the middle third point. Below this depth the general rule will apply.

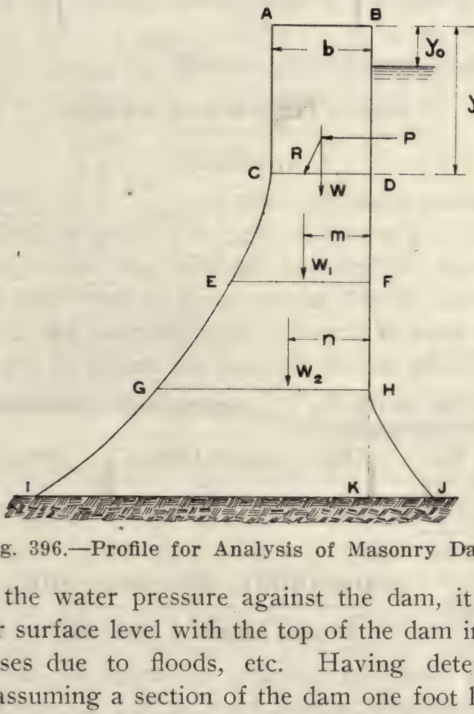

Fig. 396.—Profile for Analysis of Masonry Dams.

In computing the water pressure against the dam, it is best to consider the water surface level with the top of the dam in order to allow for possible rises due to floods, etc. Having determined the top width, b, and assuming a section of the dam one foot long, the height, y_1, of the rectangular portion can be deduced from the formula

$$y_1 = b\sqrt{s} \qquad (240)$$

in which s is the specific gravity of the material of the dam.

The down-stream face of the dam must now be sloped so as to keep the resultant pressure, with the reservoir full, at the limit of the middle third of the length of any joint. Dividing the remainder of the height of the dam into lengths convenient for computation, the length of any joint (see Fig. 396) as GH may be found by the formula

$$\overline{GH} = \sqrt{B + C^2} - C \qquad (241)$$

in which

$$(242) \qquad B = \frac{6\,m\,(\text{Area } \overline{ABFE})}{\overline{FH}} + \frac{\overline{BH}^3}{s\,\overline{FH}} + \overline{EF}^2$$

where m = distance from F of the line of action of the weight of masonry above $\overline{EF}$ and

$$(243) \qquad C = \tfrac{1}{2}\left[\frac{4\,(\text{Area } \overline{ABFE})}{\overline{FH}} + \overline{EF}\right]$$

The value of n is given by the equation

$$(244) \qquad = \frac{\text{Mom. of } \overline{ABFE} + \text{Mom. of } \overline{EFHG}}{(\text{Area } \overline{ABHG})}$$

moments being taken about the point H.

Equation (241) can be used as long as n is greater than one-third the length of the joint. When this condition can no longer be satisfied with a vertical face, it will be necessary to batter the upstream face also, so that the lines of pressure with reservoir full and empty both lie at the limits of the middle third of the length of any joint.

The length of the joints, as $\overline{IJ}$, may now be found by the formula

$$(245) \qquad \overline{IJ} = \sqrt{\frac{\overline{BK}^3}{s\,\overline{HK}} + \left(\frac{\overline{GH}}{2} + \frac{(\text{Area } \overline{ABHG})}{\overline{HK}}\right)} - \frac{(\text{Area } \overline{ABHG})}{\overline{HK}} + \frac{\overline{GH}}{2}$$

and the value of $\overline{KJ}$ is

$$(246) \qquad \overline{KJ} = \frac{2\,(\text{Area } \overline{ABHG})\quad(\overline{IJ} - 3m) - (\overline{HK} \times \overline{GH}^2)}{6\,(\text{Area } \overline{ABHG}) + \overline{HK}\,(2\overline{GH} + \overline{IJ})}$$

In high dams two more stages, governed by the compressive strength of the masonry, would have to be considered, but, within the limit of height set above, the formulas given are sufficient.

The position of the line of pressure may be readily determined also by graphical methods.

In the case of overfall dams, which are necessarily subjected to dynamic forces, which are more or less indeterminate, the design cannot be so closely figured.

303. Further Considerations.—The preceding analysis does not take into account the possibility of an upward pressure from below the dam, due to the pervious character of the foundation, or to cracks and fissures, by means of which the pressure of the head water may

be transmitted to the base of the dam. This factor is commonly ignored in dam construction but should be considered, and allowance should be made, at least partially, for upperward pressure which will be equal to the head at the upstream side of the dam and reduced to a minimum at the lower toe or at suitable drains constructed to relieve such pressure.

In most cases, the foundation must be so prepared as to reduce the upward pressure to a minimum. This may usually be done by the careful preparation of the foundation to prevent inflow, or by the construction of drains from the interior of the foundation to the lower face. In many cases a series of drainage wells connected with a drainage system should be constructed at intervals across the heel of the dam, and for dams under high heads grout may be forced into a

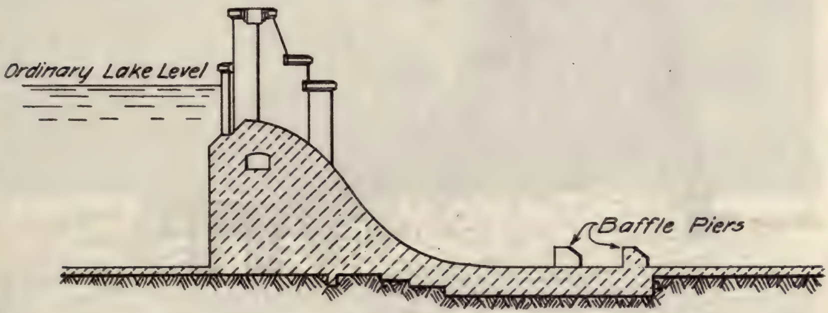

Fig. 397.—Section of Spillway of the Gatun Dam.

series of holes in the foundation rock just above the dam to advantage.

The construction of a dam with a vertical overfall, unless provision is made for the admission of air, will result in the formation of a partial vacuum below the sheet, and a certain extra resultant head on the structure due to the same. The vertical overfall is also frequently objectionable, on account of the action of the falling water on the bed of the stream immediately adjacent to the dam, and on the foundation of the dam itself. It is frequently desirable to give the lower face of the dam a curved outline, in order to guide the water smoothly over the dam, and deliver it approximately tangentially to the stream bed, and in extreme cases, a series of baffles to destroy the velocity resulting from the overfall is desirable. An example of this type of construction is shown in the Gatun dam (see Figure 397). The spillway of this dam is 760 feet long, and lies in a circular arc, directing the discharge into a concrete lined waste channel 285

feet wide. A system of baffles, consisting of two rows of obstructions with open spaces between them, was introduced at the toe of the dam.*

The convex surface of the dam should be of such form that the water will, through gravity, adhere to it. An example of a dam with a curved face is shown by Figure 398 which is a section of the dam of the Holyoke Water Power Company. Two views of the dam, one during low water (Fig. 399, page 623), and one with about ten feet of water flowing over the crest (Fig. 400, page 623) are also shown. A section of the McCall's Ferry dam, built of Cyclopean concrete (height fifty-three feet) is shown in Fig. 401, page 624.

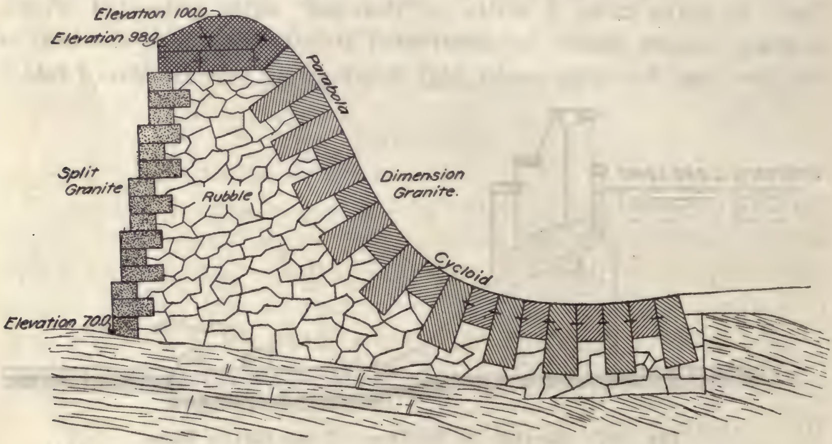

Fig. 398.—Cross-section of Dam of Holyoke Water Power Co.

Figure 402, page 625, shows a section of the reinforced concrete dam of the Wisconsin River Power Company at Prairie du Sac, Wisconsin. This dam creates a maximum head of thirty-four feet, the upper fourteen feet of which is controlled by forty-one tainter gates, each twenty feet in width. This dam is constructed on a sand and gravel foundation, and is sustained by thirty foot piles. Seepage is reduced by fifty foot interlocking steel piling, driven across the river at the upper face of the dam, and the toe is protected by thirty foot interlocking steel sheeting. Below the concrete toe a willow mattress two feet in thickness and thirty-six feet in width is placed. This mattress is held in place by a row of round piles placed close together six feet above the lower edge. The space between the apron and the piles over the mat is filled with riprap.

* See Engineering Record, June 4, 1910.

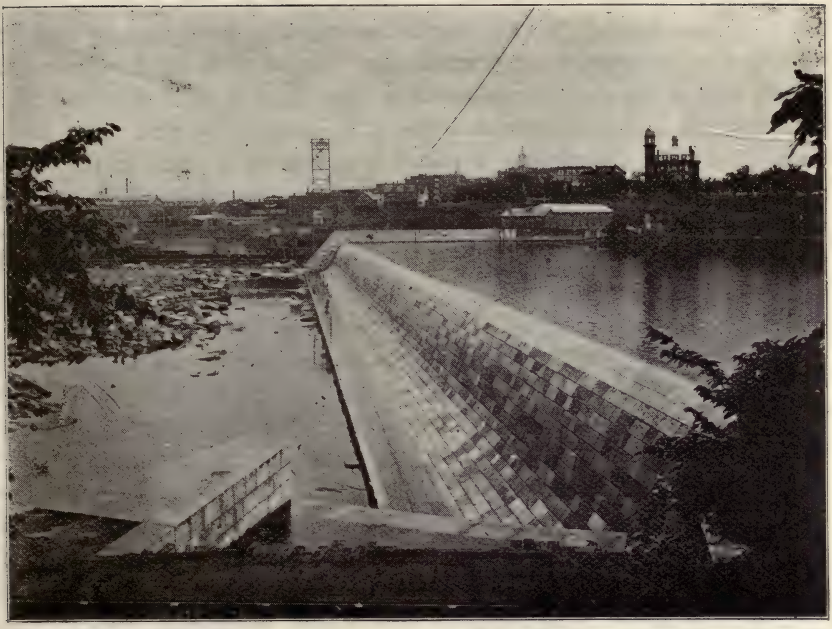

Fig. 399.—Masonry Dam of Holyoke Water Power Co. (see page 622).

Fig. 400.—Holyoke Dam During Flood (see page 622).

Figure 403, page 626, shows a cross-section of that portion of the new dam at Austin, Texas, which has recently been constructed to replace the portions of the original dam destroyed in 1900. A length of 520 feet of the old dam remains in place and is utilized in the new construction. The foundations are soft, seamy limestone and have been filled with grout; and a cutoff wall has been sunk in the limestone to such depth that it is below pervious material. The crest of the new portion is nine feet lower than that of the old dam, or fifty-one feet above low water. Flood gates fifteen feet high are carried on the crest, aggregating a total length of spillway of 1,091 feet. The gates are automatic in action, being of the leaf type, which turn about rockers when the center of pressure of water rises to a certain point.* The O. G. curve for overfall dams of this character should be kept at or above the parabolic path that the water would take in a free fall with the initial horizontal velocity corresponding to the depth of water on the dam.

Fig. 401.—Section of McCall Ferry Dam (Eng. Rec.). (See page 622.)

From equation 47, page 115, the flow over one foot of crest will equal,

$$q = vh = m\left(\tfrac{2}{3}\right)\sqrt{2g}\;h^{\frac{3}{2}}, \quad \text{hence} \tag{247}$$

$$v = m\left(\tfrac{2}{3}\right)\sqrt{2gh} \tag{248}$$

The abscissa of the parabola is $x = vt$, in which t = time in seconds. The ordinate is, $y = \frac{1}{2}gt^2$, hence

$$y = \frac{g}{2y^2}\,x^2 \tag{249}$$

is the equation of the parabola.*

When a curved face is impracticable or undesirable and the bed of the stream, below the dam, is not of suitable material to resist the impact of the falling water, some form of apron must be provided. Sometimes the dam is divided into steps over which the water falls in numerous cascades. Such a dam is shown in Fig. 404, page 627.

* Engineering Record, May 29, 1915.

* Turneaure & Russell's "Public Water Supplies," Section 446.

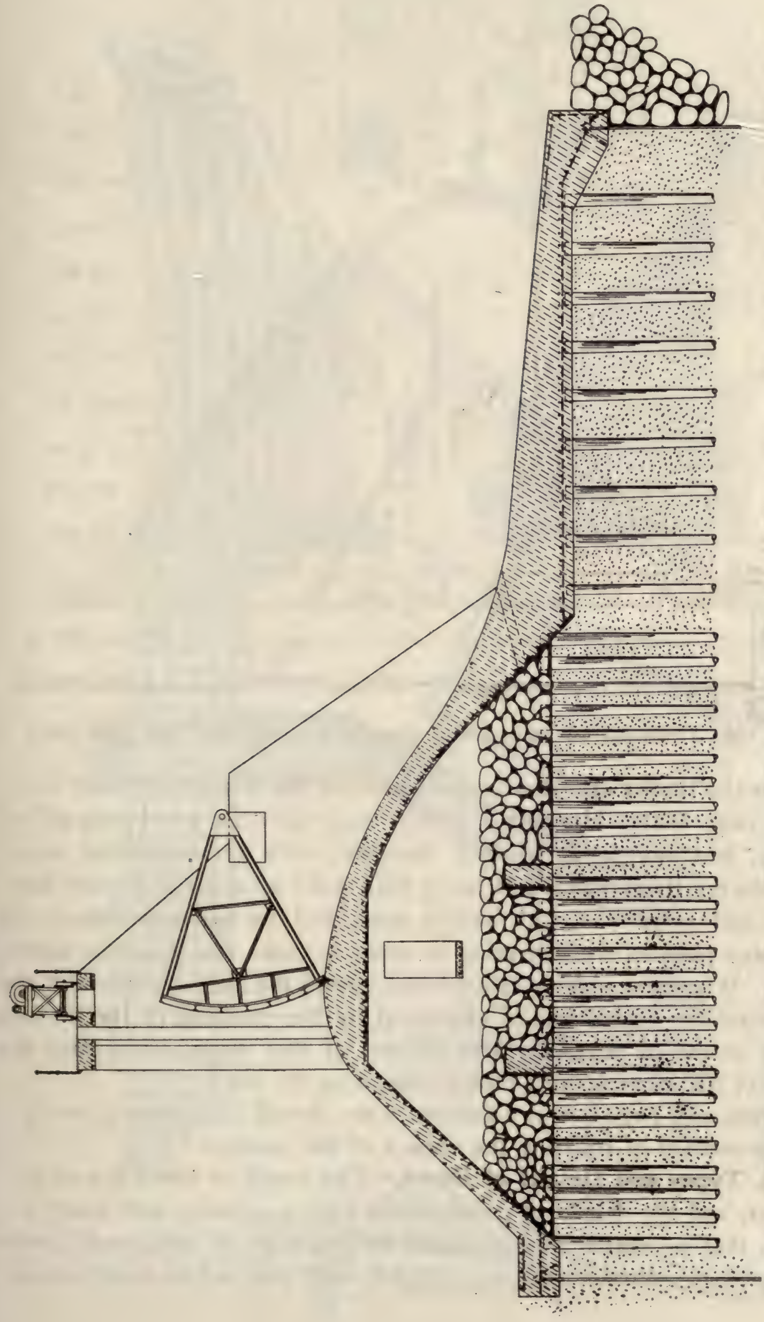

Fig. 402.—Cross-section of Reinforced Concrete Dam of the Wisconsin River Power Co. at Prairie du Sac, Wis. (see page 622).

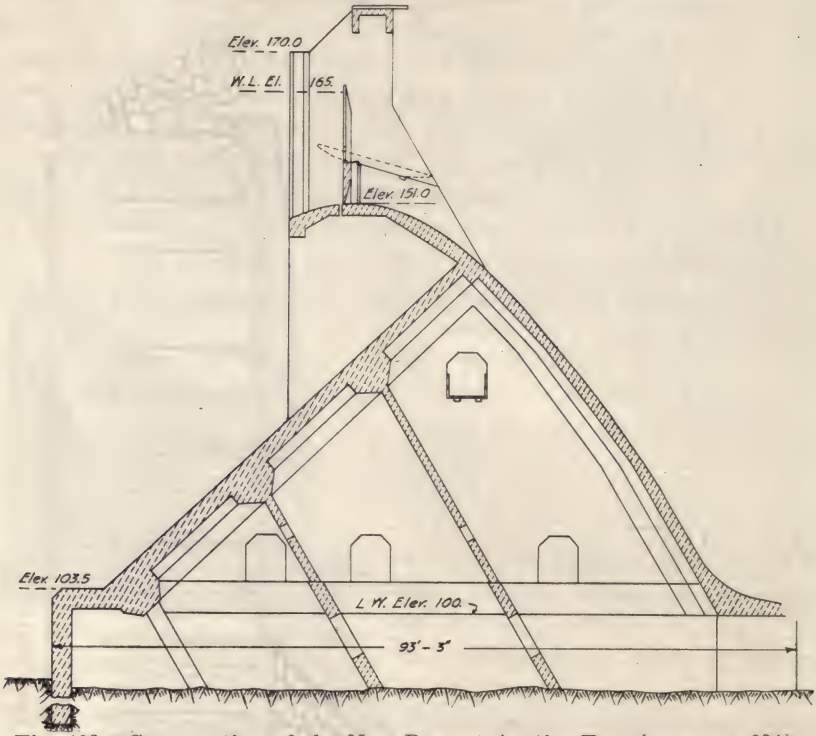

Fig. 403.—Cross-section of the New Dam at Austin, Tex. (see page 624).

This is the timber crib dam constructed for the Montana Power Company, near Butte, Montana. In this case the cells are composed of timber, laid alternately in each direction, with a considerable space left between them, instead of being built solid as in the Kilbourn dam. These cells were filled with broken stone, and the upstream side of the dam was planked with sheeting in order to make the structure watertight. When the water was admitted behind the dam a portion of the structure was forced out of alignment by the crushing of the timbers at the points of contact. The amount of this displacement and the cause of the same is quite clearly shown in the cut.

Figure 405, page 628, is a section of the Sewall Falls dam, showing a similar method of resisting the impact of the overflow.

304. Types and Details of Dams.—The types of dams are so numerous, and the details of construction vary so greatly with every locality, that an entire volume would be necessary to adequately cover this subject. As the subject is already well covered in many special

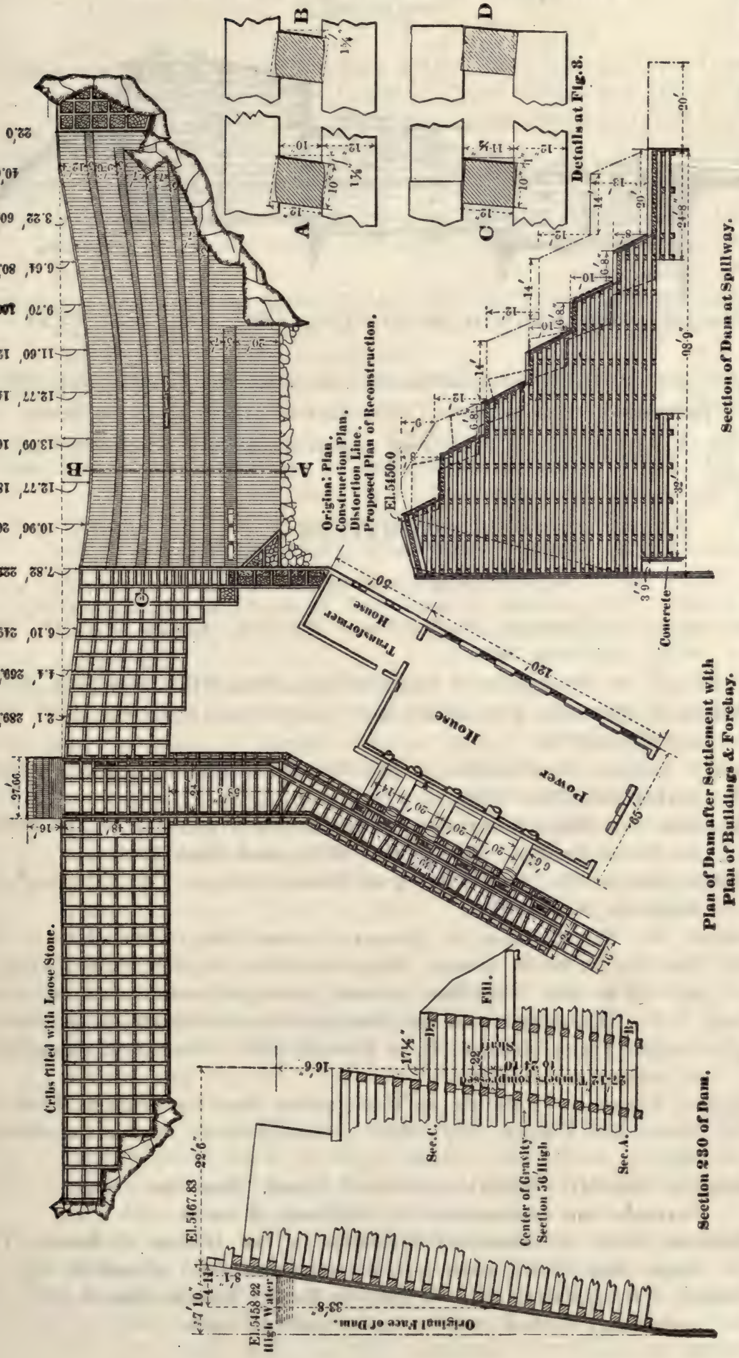

Fig. 404.—Timber Crib Dam of the Montana Power Co., Near Butte, Montana (Jour. Asso. Eng. Soc., Vol. 22, page 175).
(See page 624.)

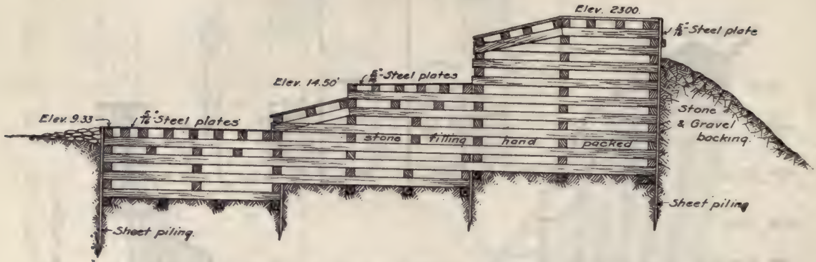

Fig. 405.—Timber Dam at Sewall Falls. (Eng. News, Vol. XXXI.)

treatises and articles, no attempt will be made to discuss this subject in the present edition. Numerous references are given to books and articles in which special forms of construction are discussed and described.

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CHAPTER XXI

APPENDAGES TO DAMS

305. **Movable Dams.**—The height of a dam is limited in the manner hereinbefore described. It will be noted that the limit is that imposed by high water conditions and that, as a rule, the water surface during low stages could be raised to a considerable amount without interference with the riparian owners, if at the same time flood conditions could be provided for. In order to provide such conditions, movable dams are sometimes constructed which will permit of raising or lowering all or a part of the structure as the stage of the water

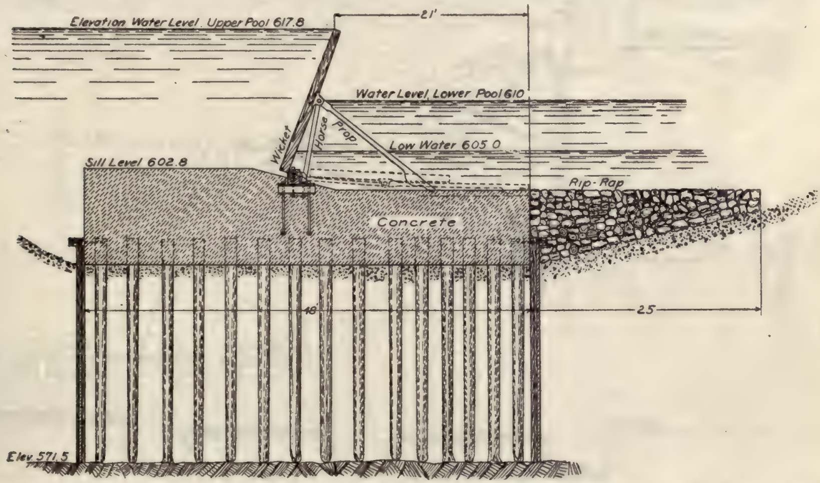

Fig. 406.—U. S. Movable Dam on Pile Foundation, McMechen, W. Va. (Eng. News, Vol. 54, page 100).

requires. These flexible portions of the dam may consist of a gate or series of gates which can be raised or lowered. Sometimes a considerable portion of the dam is made flexible by the construction of a bear trap leaf, which is usually raised and lowered by hydraulic pressure, and by means of which the head of water can be readily and rapidly controlled. Sometimes the entire dam is made movable by the use of Chanoine wickets (see Figure 406) and similar types of dams, a part of which may be removable while other parts are folded down on the bed of the stream, allowing the flood waters to

pass over them. Most of such constructions are expensive and are used most largely on government works for the control of rivers for navigation purposes.

The objection to movable dams for water power purposes is that the reduction in the elevation of the head water by their use commonly so reduces or destroys the head that the continuity of the power output is interrupted. The same objection also applies to any gate, flash board or other device designed to reduce the head. Such reduction

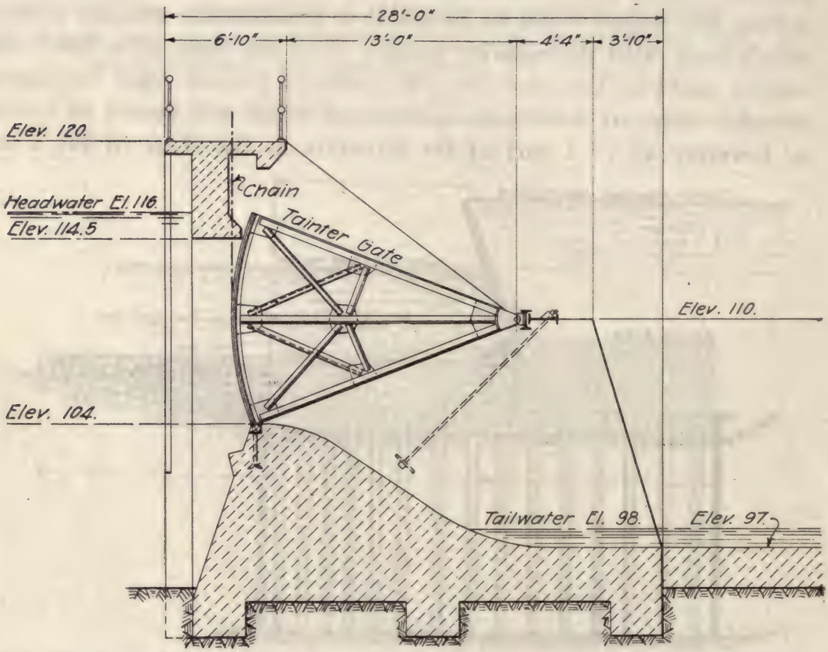

Fig. 407.—Tainter Gate Section of the Alpena Power Co. (see page 649).

is usually made during conditions of flow under which the natural head that would obtain is already at a minimum.

306. Flood Gates.—Flood gates are quite commonly used in water power dams to control or modify extreme flood heights. These gates are commonly designed to be raised so as to permit the escape of the water underneath them. The tainter gate, in some of its modifications, is perhaps most widely used for this purpose. Fig. 390, page 613, shows a cross-section of the tainter gate dam constructed to control the flood height of the Black River at Black River Falls, Wisconsin, and which replaced the solid section of limited spillway

capacity which was one of the causes of the heavy flood loss to that city in the October flood of 1911.

Figure 407, page 648, is a section of one of the six gates designed for the dam of the Alpena Power Company at Alpena, Michigan. These gates are operated by a movable hoist, similar to Fig. 408 which travels on a track on the bridge above.

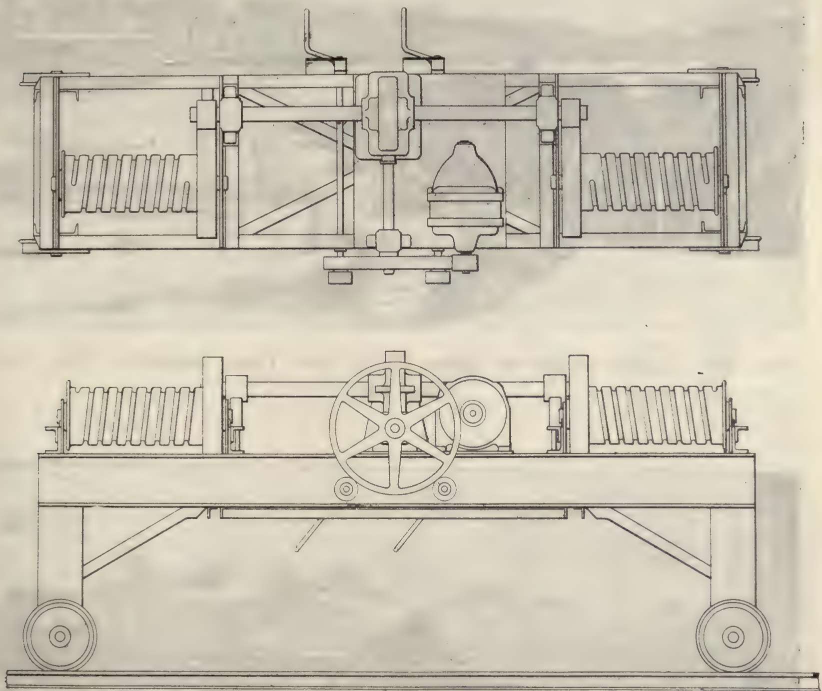

Fig. 408.—Movable Tainter Gate Hoist.

Figures 409 and 410, page 650, are views of the steel tainter gates constructed in the upper and lower United States Government dams across the Fox River at Appleton, Wisconsin.

In the dam of the Southern Wisconsin Power Company at Kilbourn, Wisconsin, the rise of the flood water is so great (about sixteen feet) that it was found impracticable to construct lift gates to reduce the flood heights. In this case the crest was divided by piers, into twelve twenty-five foot sections. Between each two piers a twenty-five foot gate is placed (see Fig. 411, page 651) which can be lowered into the dam six feet, thus reducing the extreme flood height by that amount. These gates are of steel and weigh about seven tons each. They may

Fig. 409.—Tainter Gates at Upper U. S. Gov. Dam, Appleton, Wis. (see page 648).

Fig. 410.—Tainter Gates at Lower U. S. Gov. Dam, Appleton, Wis. (see page 648).

be operated by an electric motor or may be manipulated by hand, should occasion require.

Under some circumstances the use of gates across the crest of a dam is not without danger. Care must be taken to see that ice or

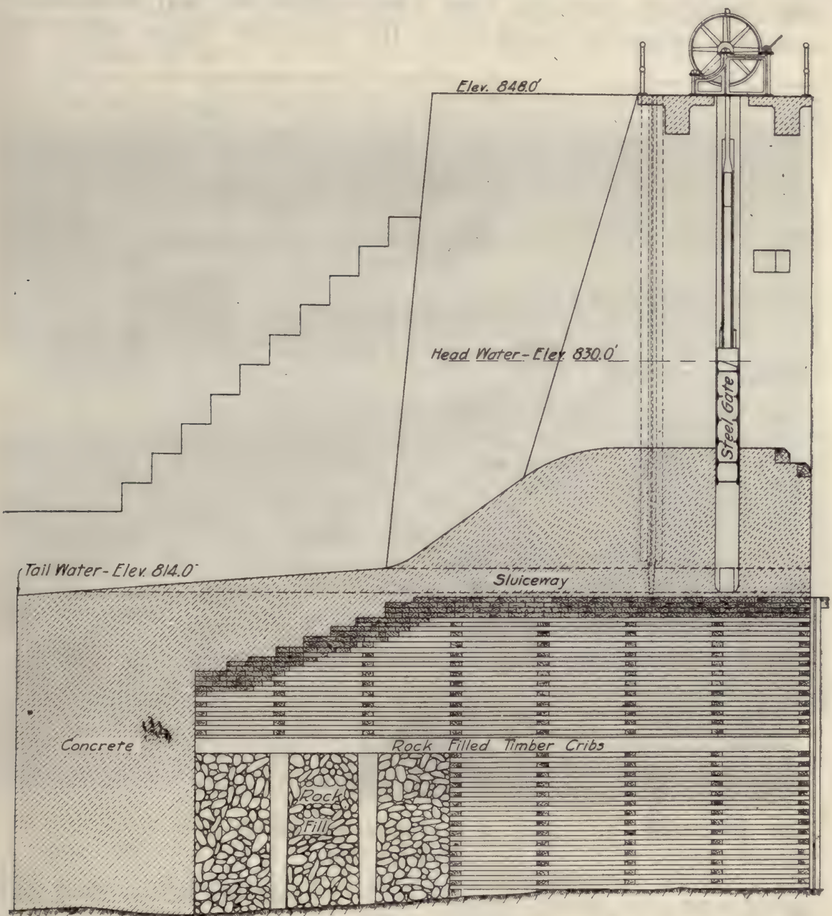

Fig. 411.—Dam at Kilbourn, Wis., With Movable Crest, Southern Wisconsin Power Co. (see page 648).

floating material which may lodge against the gate piers is promptly removed and the opening kept unobstructed. In general, gates should be avoided except as an auxiliary means of flood control in rivers used largely for the floating of logs, as an unusual flood may bring down such a mass of logs as to be beyond control, and effectively shut off

the gate area. Such an occurrence resulted in a heavy loss at the Rothschild dam on the Wisconsin River in the October flood of 1911. The tainter gates were kept closed in order not to endanger the apron, which was in poor condition and undergoing reconstruction (see Fig. 395, page 618), until the flood waters above the dam threatened to seriously injure the property of the Marathon Paper Mills. The

Fig. 412.—Rothschild Dam of the Marathon Paper Mills During Flood of October, 1911.

gates were then rapidly opened, and the rapid fall of the river above the dam resulted in the floating boom which empounded a large number of logs just above the dam, catching on the boom piers and allowing the logs to escape. These logs effectively clogged the gates, and to protect the mill property a cut was started around the opposite side of the dam which was rapidly enlarged by the river, as shown in Figure 412. This new channel has since been closed at a large expense, the new dam being about the same length as the dam originally constructed.

307. **Flashboards.**—The control of limited variations in head is commonly accomplished by means of flashboards which are widely used for this purpose. The simplest form of flashboard consists of a line of boards placed on the crest of the dam (see Fig. 413) usually held in place by iron pins to which the boards are commonly attached by staples. The object of flashboards is principally to afford a certain pondage to carry the surplus water from the time of minimum

Fig. 413.—Flashboards and Supports, Rockford Water Power Co.

use of power to the time of maximum demand. Incidentally, the head is raised and the power is also increased in this way. The supports of the flashboards should be so arranged that they will withstand only a comparatively low head of water flowing over the boards, and will be carried away if a sudden flood should raise the head materially above a safe elevation. If the boards are so supported as to withstand the discharge of heavy floods, they will form a permanent portion of the dam and increase its fixed elevation to such an extent as to create damage which their use is supposed to avoid. Sometimes the pins supporting the boards are made so light that they must be held in posi-

tion by inclined braces. These braces are sometimes supplied with steel eye-bolts through which is passed a cable. A large steel washer is attached at one end and a winding drum at the other (see Fig. 413, page 653). Commonly, if a flood is anticipated, the boards are removed and stored for future use. If, however, a sudden flood should arise, the inclined braces are removed by winding up the cable and the pressure on the flashboards bends the pins and the boards are washed away. The expense involved by the loss of flashboards is not excessive as one set will commonly take care of the entire summer low water

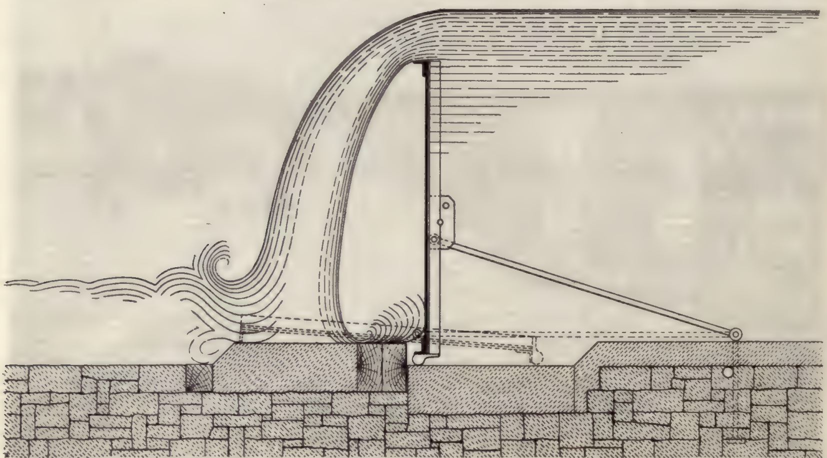

Fig. 414.—Automatic Drop-Shutter for Betiva Dam, India (Eng. News, June 4, 1903).

period. The expense involved in their use is therefore only the cost of one set of flashboards per year.

Sometimes the flashboards constitute a permanent but adjustable part of the dam and are lowered automatically during stages of high water (see Fig. 414; also Fig. 415, page 655). On some dams, especially at waste weirs of canals and reservoirs where the fluctuations in height are inconsiderable, the dam may be provided with a foot bridge which makes the whole crest of the dam accessible at all times and from which the flashboards can be readily adjusted. This plan is used on the dam across the Black River at Hatfield, Wisconsin (see Fig. 416, page 656) which is however also provided with a large tainter gate section. Ordinarily, on rivers subject to high floods, this type of construction is impracticable.

The Pennsylvania Water Power Company has installed in its McCall Ferry dam a system of flashboards which permits them to hold the head water to a maximum of four feet six inches above the concrete crest of the dam, gaining thereby additional head under ordinary flow and additional storage for use during low flow.

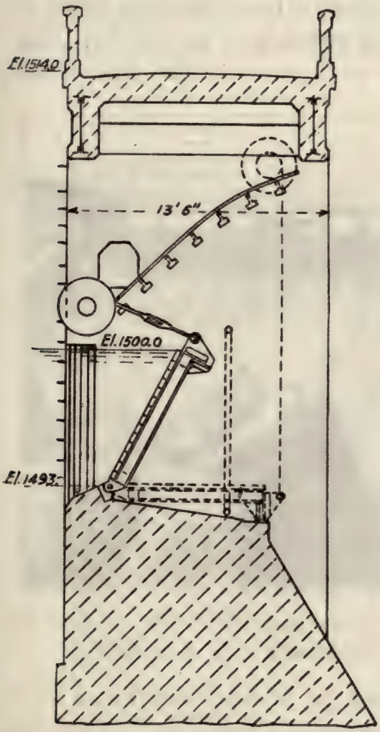

Fig. 415.—Automatic Flashboards on the Dam at Tallulah Falls, Ga. (see page 654).

The flashboards are built of one and one-half inch plank, in sections 4′6″ x 16′, and are supported by vertical steel pins set in pipe-bushed holes, spaced two feet six inches, in the crest of the dam.

The pins are so dimensioned that for a rise in head water of nine inches above the tops of the flashboards, a section of 500 feet in length would be carried away and a further rise of six inches would carry away the remaining 1,900 feet. For better determined and more uniform action, the pins were reduced in diameter by a circumferential groove two inches above the crest of the dam, the diameter at this point being 2 5/16″ x 2⅛″ for the 1,900 feet and 500 feet sections, respectively. The flashboards are handled from a barge equipped with a steam operated derrick.

A row of guard pins is installed on a line eighteen inches upstream from the flashboard pins, spaced seven feet six inches.

In some dams, instead of gates or flashboards, vertical stop planks or needles are used. These consist of planks or squared timbers that are lowered vertically into position, stopping off the opening partially or wholly, as desired. They are commonly supported by a shoulder at the bottom of the opening and one or more cross beams above.

308. Head Gates and Head Gate Hoists.—It is usually desirable to control the water at the inlets to the head-race flume or pipe line

by the use of gates which may be closed in emergencies or for the purpose of making necessary repairs or modifications in the raceway through which the water is diverted to the plant. In northern rivers it is also found desirable to prevent the entrance of ice into the raceway either by the construction of a floating or fixed boom in front of the gates or by constructing a system of submerged arches (see Fig. 359, page 566) either in front of, or as a part of, the gate ways.

Fig. 416.—Dam of the Wisconsin Railway, Light and Power Co. at Hatfield, Wis., Showing the Adjustable Flashboard (see page 654).

By means of such construction the floating ice or other floating material may be diverted from the raceway and passed over the spillway of the dam.

The head gates must be sufficiently substantial to allow the race to be emptied under ordinary conditions of water and to protect the raceway under flood conditions.

Figure 417, page 657, shows the details of the head gates designed for the power plant at Constantine, Michigan. A rear view of these gates from the race side is also shown in Fig. 418, page 658. These gates are double wooden gates with concrete gateways and are arched over between the piers so as to permit the passage of men and teams. These gates are designed to pass about 2,000 cubic feet per second.

Figure 419, page 659, shows a set of double wooden gates, the posts and braces of which are made of structural steel designed for the power plant of Mr. Wait Talcott, at Rockford, Illinois.

In the Constantine gates the gate mechanism is geared for fairly rapid operation by two men. The Rockford gate apparatus is very simple, the gate being handled with a capstan bar by a single man but at a much slower speed.

Figure 420, page 660, shows the movable head gate hoist designed for the operation of the head gates at the Kilbourn plant of the South-

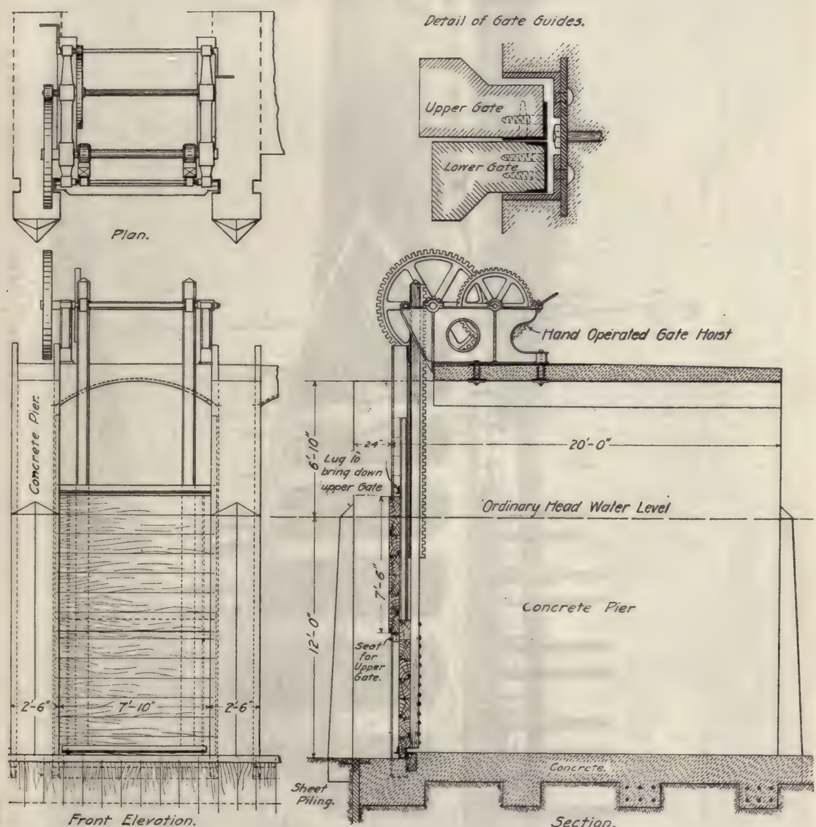

Fig. 417.—Details of Head Gates at Constantine, Mich. (see page 656).

ern Wisconsin Power Company. This hoist is operated either by hand or by an electric motor not shown in the cut.

309. Fishways.—In almost every state fishways are required by law in any dam constructed on natural waterways. These fishways must be so arranged as to permit the free passage of fish up the stream. Their actual use by fish is however comparatively rare and their prac-

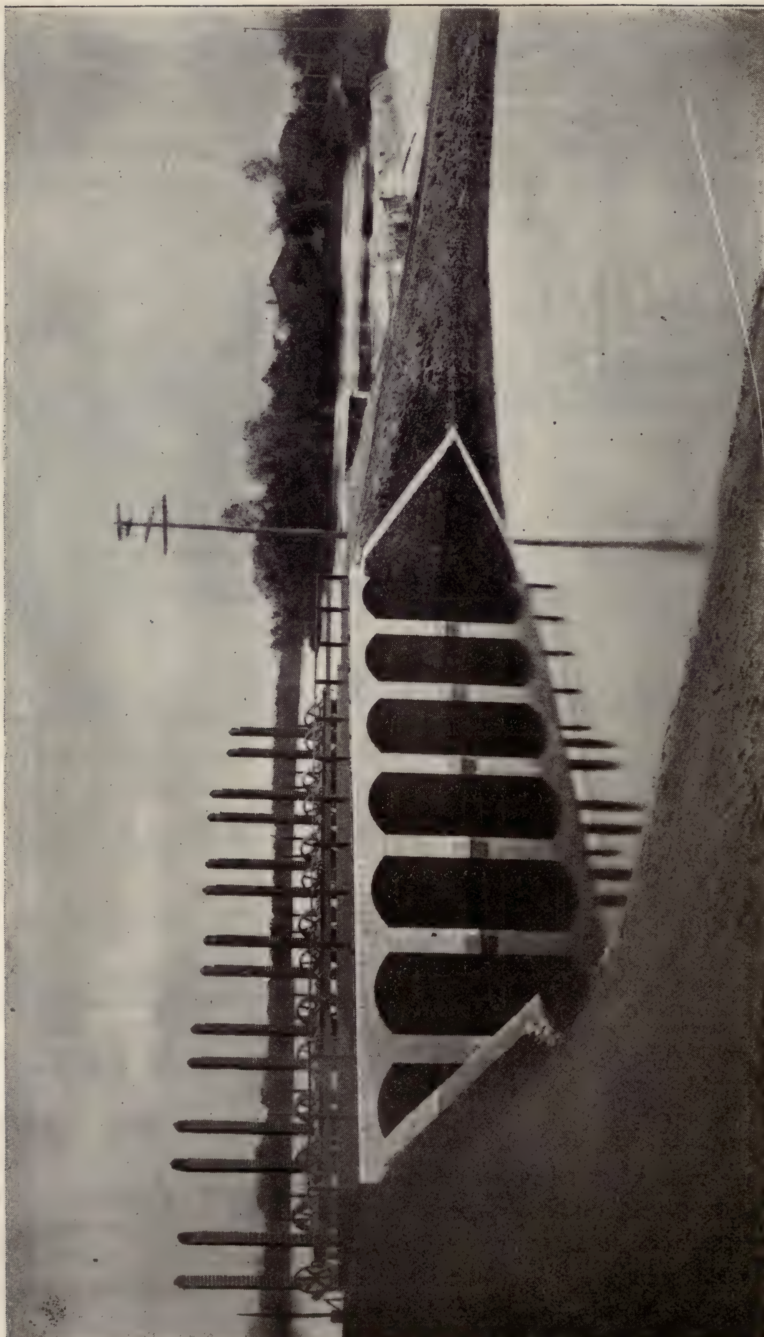

Fig. 418.—Rear View of Head Gates at Constantine, Mich. (see page 656).

tical utility is doubtful. Fig. 421, page 661, is a fishway designed by Mr. L. L. Wheeler and constructed in the dam at Sterling, Illinois. The Sterling dam is a timber crib dam and the fishway is constructed

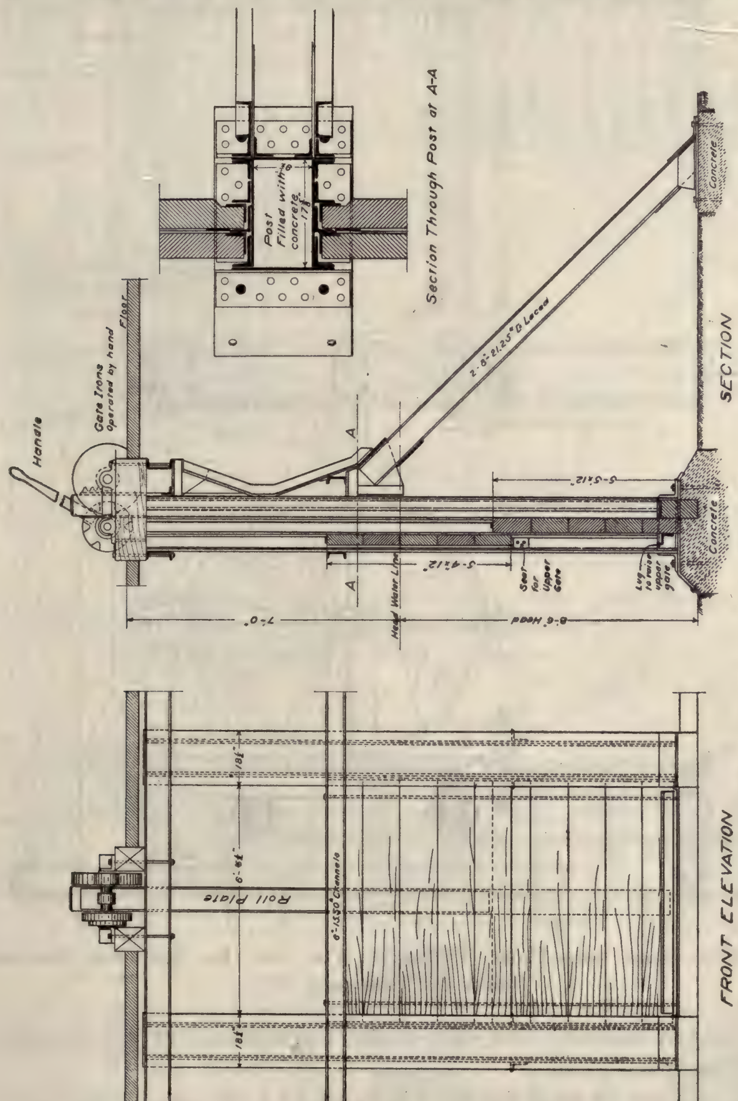

Fig. 419.—Details of Head Gates for Mr. Wait Talcott, Rockford, Ill. (see page 656).

of timber. Fig. 422, page 661, shows the type of fishway recommended by the Fish Commission of the State of Wisconsin and ordinarily used in that state. The middle diagram in Fig. 425, page 664, shows a fish-

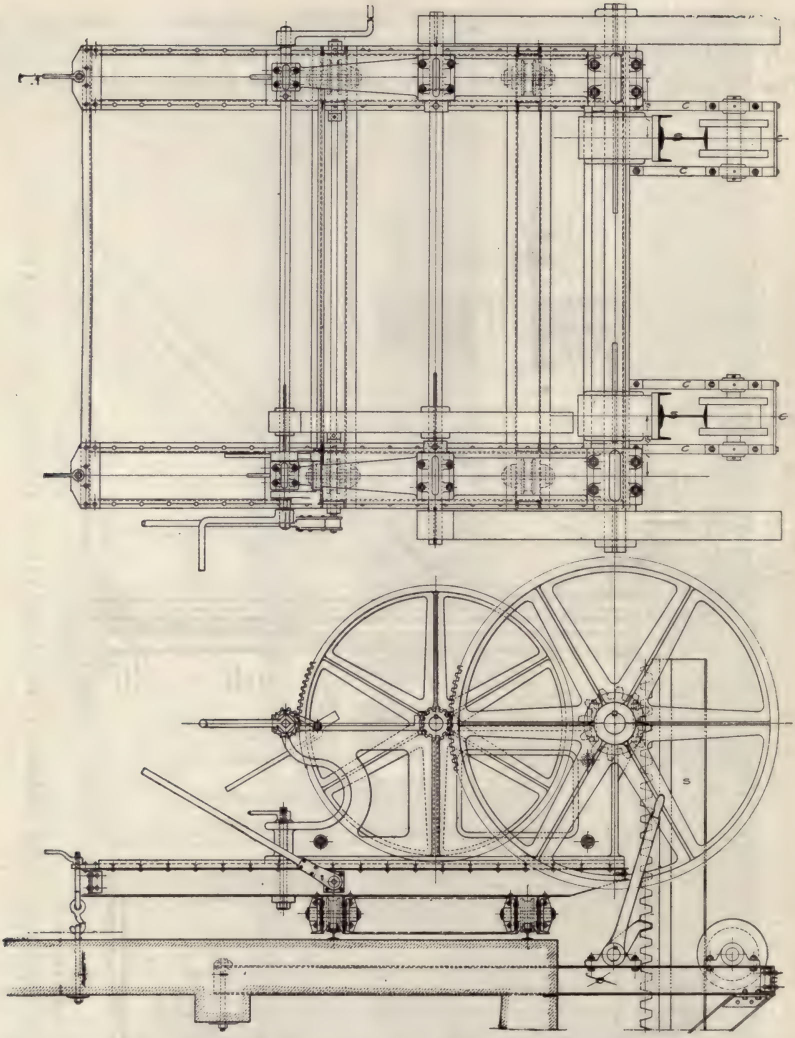

Fig. 420.—Head Gate Hoist, Kilbourn, Wis., for the Southern Wisconsin Power Co. (see page 657).

way used in the forty foot dam of the Peninsular Power Company on the Menominee River.

The purpose of these fishways is to afford a gradual incline through which a continuous stream of water of comparatively low velocity shall flow and against which the fish may readily swim. Both the inlet and outlet should be below low water and the outlet should be in such a

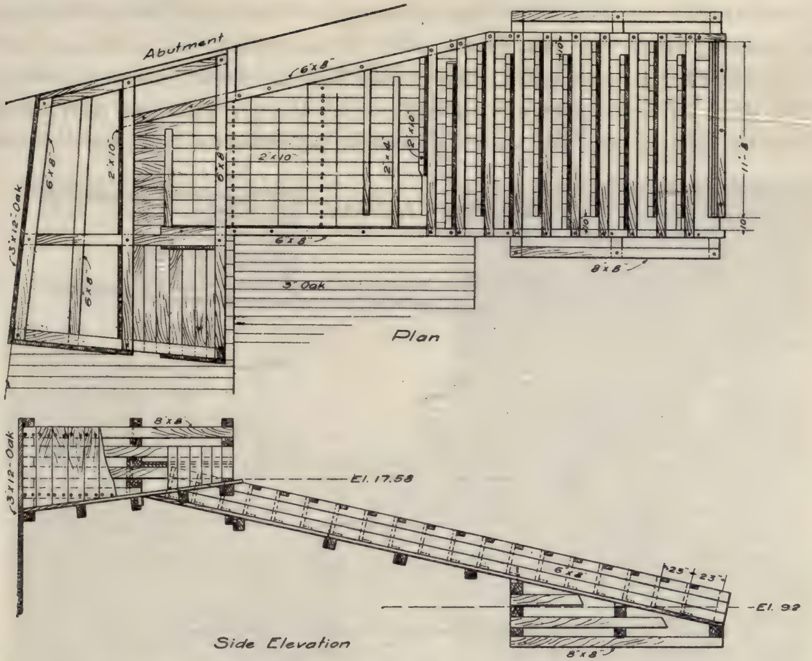

Fig. 421.—Timber Fishway in Dam at Sterling, Ill. (Eng. News). (See page 659.)

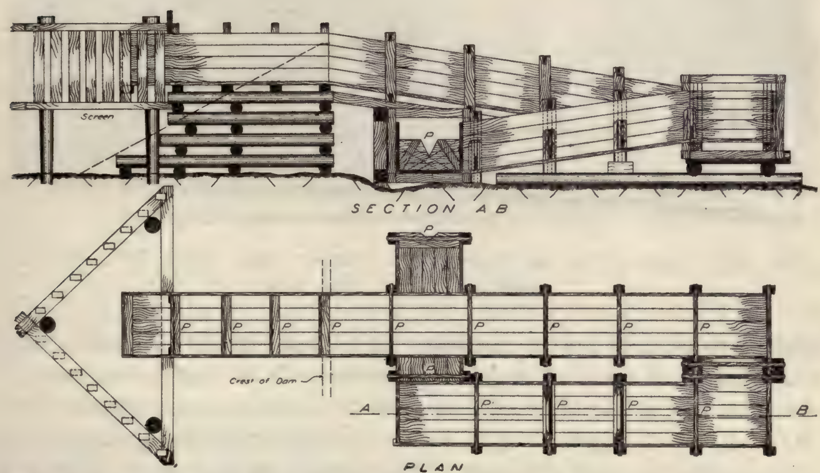

Fig. 422.—Fishway of Fish Commission, State of Wisconsin (see page 659).

position that the fish, when they ascend the stream and reach the dam, in passing from one side to the other in searching for a passage, are naturally led to the point where the flowing water is encountered. The slope of these fishways should not be steeper than one vertical to four horizontal, and the water should be so deflected that the velocity

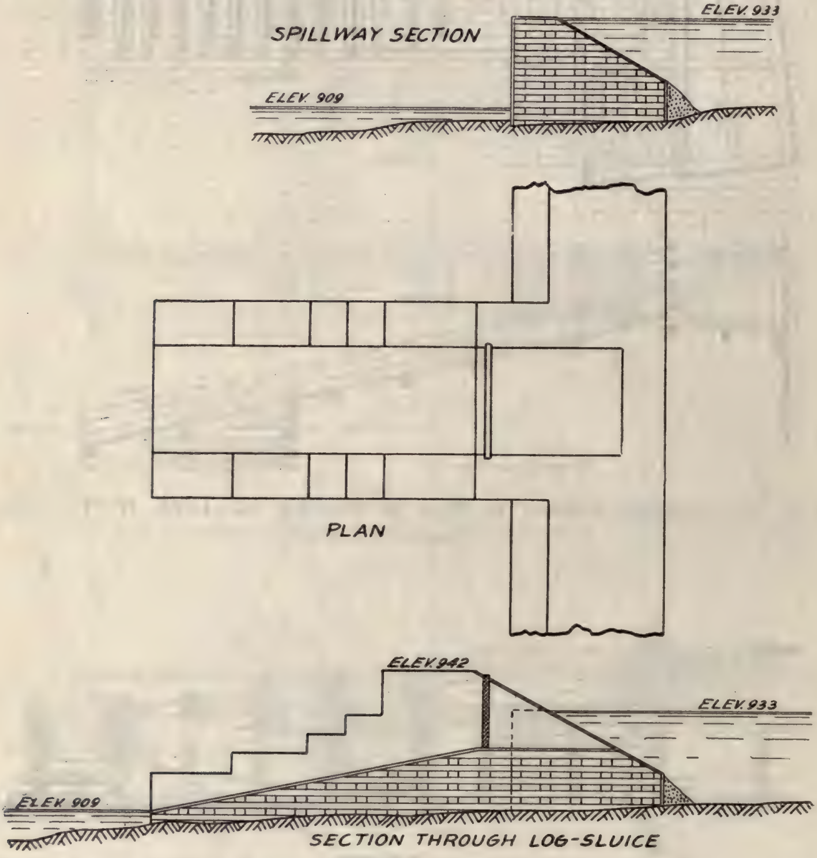

Fig. 423.—Logway in the Chesuncook Timber Dam (Eng. Rec., Vol. 50, page 70). (See page 663.)

will be reduced as low as possible. A fishway should be entirely automatic and free from all regulating devices. It is usually desirable for the openings in the bulkheads or baffles to increase progressively from the lower to the upper one in order to insure that the passage of the fishway shall be full of water. The fishway should be so covered as

to prevent interference, but must be light or it will not be used by the fish.

310. Logways.—The free navigation of streams for logging purposes is provided by law in most states and it is therefore necessary where logging is practiced to provide ready means for their passage

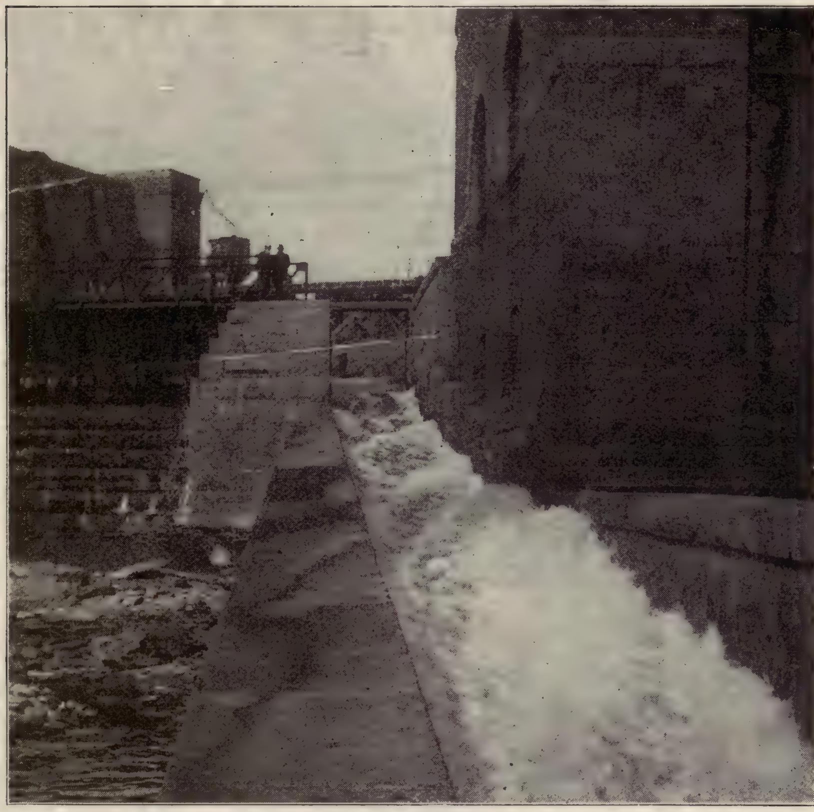

Fig. 424.—Logway at Lower Dam, Minneapolis, Minn.

over or through the dam. Formerly provision had to be made for the passage of rafts but in most cases this is unnecessary at the present time. The passage of logs is accomplished in the Kilbourn dam (see Fig. 411, page 651) by the lowering of any one of the flood gates.

Figure 423, page 662, shows a plan and section of the log-sluice constructed in the Chesuncook timber dam on the Penobscot River. A section of the spillway of the dam is also shown in the same figure.

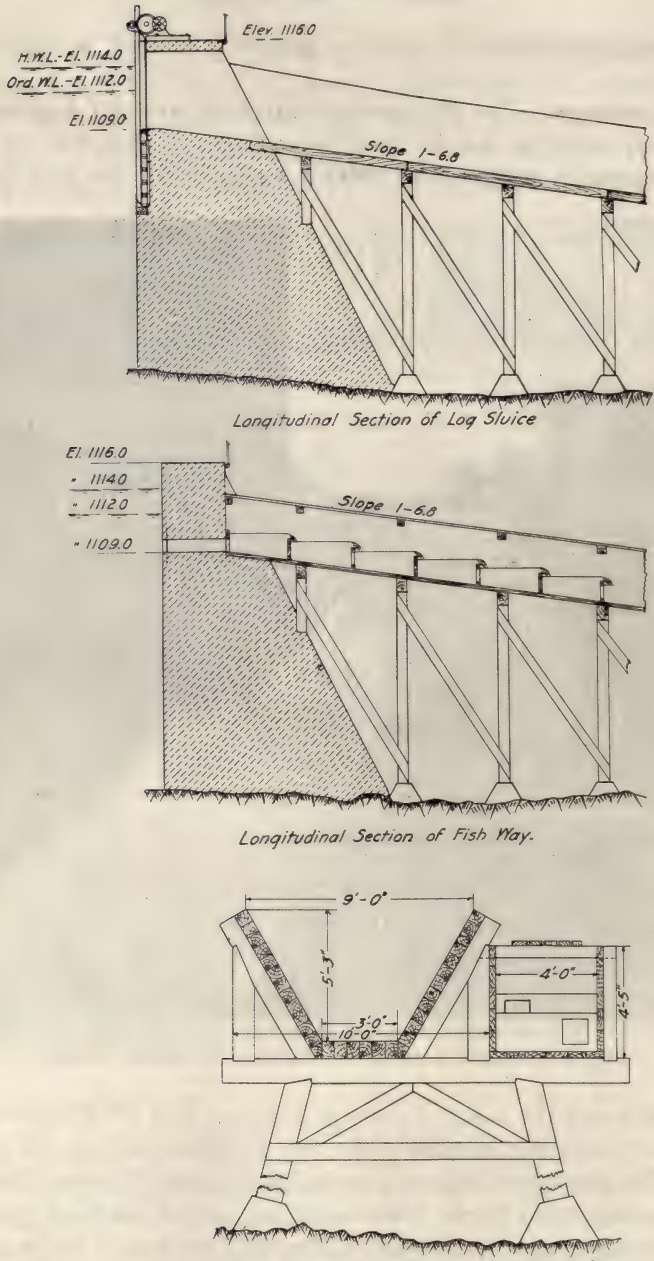

Fig. 425.—Sections Through Log-Sluice and Fishway in the Twin Falls Dam of the Peninsular Power Co. (see pages 659 and 665).

Figure 424, page 663, is a view of the logway in the lower dam at Minneapolis. This sluice is only six or eight feet in width, and the depth and quantity of water flowing is controlled by a bear trap leaf.

Figure 425, page 664, shows the log-sluice constructed of timber in the dam of the Peninsular Power Company.

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CHAPTER XXII

COST OF POWER PLANTS AND OF POWER

311. Cost of Water Power Development.—The various conditions under which water power is developed greatly affect the cost of development. As a general rule, other things being comparatively equal, the larger the power developed the smaller the cost of development per unit capacity. This is particularly true when developments of various capacities are considered on the same stream. Many of the features of the development must be essentially the same regardless of the ultimate capacity of the plant. This is especially true of dams and river protection work. The variation in cost per unit capacity of various sized plants is illustrated by Table 39.

TABLE 39.

*Estimate of the Cost of a Hydro-Electric Plant at Niagara Falls.**

Items.	24-Hour Power Capacity.		
	50,000 H. P. Development.	75,000 H. P. Development.	100,000 H. P. Development.
Tunnel tail-race	$1,250,000	$1,250,000	$1,250,000
Headworks and canal	450,000	450,000	450,000
Wheel pit	500,000	700,000	700,000
Power house	300,000	450,000	600,000
Hydraulic equipment	1,080,000	1,440,000	1,980,000
Electric equipment	760,000	910,000	1,400,000
Transformer station and equipment	350,000	525,000	700,000
Office building and machine shop	100,000	100,000	100,000
Miscellaneous	75,000	75,000	75,000
	$4,865,000	$5,900,000	$7,255,000
Engineering and contingencies 10 per cent.	485,000	590,000	725,000
	$5,350,000	$6,490,000	$7,980,000
Interest, 2 years at 4 per cent.	436,560	529,584	651,168
Total capital cost	$5,786,560	$7,019,584	$8,631,168
Per horse power	$114	$94	$86

* First report of Hydro-Electric Power Commission of the Province of Ontario, page 15.

Other things being comparatively equal, the cost of development varies inversely, although not in the same ratio, as the head. The reason of this is evident from the fact that while the power of a stream is directly proportional to the head, the power output of a turbine increases as the three-halves power of the head. With double the head the power of a wheel is increased almost three times.

For moderate changes in head, the cost of the turbines will vary in proportion to their size and not their capacity; so that the cost per unit of capacity will usually decrease considerably with the head. The cost per unit of capacity of other features of water power plants will also frequently decrease as the head increases. This is particularly true of pondage capacity which increases in value directly as the head increases, although the cost per unit of land overflowed may remain constant. The relative cost of high and low head developments may be illustrated by the comparative cost of two plants recently designed which were of approximately the same capacity but working under different heads. The comparison is as follows:

TABLE 40.

Comparative Cost of Water Power Plants.

Capacity.	Head.	Cost of Water Power Development.			
		Without dam.	With dam.	With dam and electrical equipment.	With dam, electrical equipment and transmission line.
8,000	18	63.50	86	115	150
8,000	80	21.	39	60	90

The estimates of The Ontario Hydro-Electric Power Commission of the cost of various hydro-electric plants proposed in Ontario, furnish a good example of the variations in the cost, per unit of power, of various plants under various conditions. These estimates are shown in Table 41, page 670.

The actual costs per horse power capacity of various complete American and foreign plants are shown in Tables 42, page 671, and 43, page 672, respectively.

312. **Cost of Water Power.**—The cost of water power depends on:

First: Cost of financing including discount on stocks and bonds, interest during construction, cost of management and engineering, and fixed and operating charges until the plant shall reach a paying basis.

Second: The investment in real estate, water rights, power plant and equipment, transmission lines, sub-stations, distribution system, and other physical features, and the interest which must be paid thereon.

Third: On the loss from the depreciation of the various elements of the plant, the cost of maintenance and repairs, the cost of contingent damages from floods or other accidents.

TABLE 41.

Estimates of the Cost of Developing Various Canadian Powers From Reports of Ontario Hydro-Electric Power Commission.

Location of Proposed Development.	Natural head.	Available head.	Power developed, H. P.	Estimated capital cost.	Cost per H. P.
(1) Healey's Falls, Lower Trent River..	....	60	8,000	$675,000	$ 84.38
Middle Falls, Lower Trent River....	....	30	5,200	475,000	91.37
Rauney's Fall	....	35	6,000	425,000	69.67
Rapids above Glen Miller............	....	18	3,200	350,000	109.38
Rapids above Trenton................	....	18	3,200	370,000	115.63
(2) Maitland River	....	80 (5)	1,600	325,000	203.12
Saugeen River	....	40	1,333	250,000	187.53
Beaver River (Eugenia Falls)......	....	420	2,267	291,000	128.28
Severn River (Big Chute)...........	....	52 (6)	4,000	350,000	87.50
South River	....	85	750	115,000	153.33
(3) St. Lawrence River, Iroquois, Ont.	....	12	1,200	179,000	149.16
Mississippi River, High Falls, Ont. A	....	78 (7)	2,400	195,000	81.25
Mississippi River, High Falls, Ont. B	....	78	1,100	123,000	181.82
Montreal River, Fountain Falls, Ont.	....	27	2,400	214,000	89.16
(4) Dog Lake, Kaministiquia River......	347	310 (8)	13,676	832,000	61.00
	347	310	6,840	619,700	91.00
Cameron Rapids	39	....	16,350	815,000	50.00
	39	....	8,250	600,000	73.00
Slate Falls	31	40	3,686	357,600	97.00
	31	40	1,843	260,000	141.00

Third Report; (5) Dam rather expensive. (6) Head works and canal less expensive than ordinary. (7) With storage developed. (8) Including 3,500 feet of head water tunnel.

Fourth: The operating expenses, including labor, oil, waste, and other station supplies and expenses, including also, in hydro-electric plants, the patrolling and maintenance of the transmission lines and distribution system.

Fifth: The expenses for taxes, insurance, etc.

The total annual cost due to the above sources of expense is the annual cost of the power furnished by the plant, be the quantity of that power much or little.

The investment charge should be liberally estimated and should include the entire expense of development as above outlined including cost of installing and operating any auxiliary power plant needed. All contingencies should be carefully and liberally estimated. A serious

TABLE 42.

Development Costs of Various American Water Power Plants (see page 669).

Name or location of plant.	Reference.	Head in feet.	Horse power capacity at turbine shaft.	Cost.	Cost per H. P.	Notes see page 672.
1. Chicago Drainage Canal, Lockport, Ill.	Elec. World, 1906, Vol. 47, page 398	28	15,500	$3,500,000	$225.80	d
2. Columbus, Ga.	Elec. World and Engr., 1904, Vol. 43, page 165	40	9,000	450,000	50.00	c and e
3. Catawba, S. C.	Eng. Record, 1904, Vol. 50, pages 114, 129	25	10,000	1,100,000	110.00	d and f
4. Tariffville, Conn.	Amer. Electrician, 1900, Vol. 12, page 107	31	2,300	....	125.00	d
5. Delta, Penna.	Eng. News, 1898, Vol. 39, page 250	42	550	30,000	54.00	d and g
6. Lachiné, Montreal	Elec. World, 1898, Vol. 31, page 744	16	6,600	957,200	145.80	d and h
7. Winnipeg, Manitoba	Elec. World, 1906, Vol. 47, page 1291	40	25,600	4,000,000	156.25	d and i
8. Manchester, N. H.		30	6,000	....	66.00	a and j
9. Lowell, Mass.		13	....	....	110.00	a and j
10. Lowell, Mass.	The Engineer, 1902, Vol. 39, page 64	18	....	....	57.00	a and j
11. Big Cottonwood, Utah		370	3,000	325,000	108.25	d and k
12. Lawrence, Mass.		.....	1,000	....	67.50	a and j
13. Spier Falls, N. Y.	Scien. Amer. Sept. 12, 1903	90	50,000	2,100,000	42.00	c

TABLE 43. *Development Costs of Various Foreign Water Power Plants* *(see page 669).*

Name or location of plant.	Reference.	Head in feet.	Horse power capacity at turbine shaft.	Cost.	Cost per H. P.	Notes see below.
Zürich, Switzerland......	Electricity (N. Y.), 1899, Vol. 16, p. 148	Very low.	25,300	$4,650,000	$183.90	d and l
Rhinefelden, Germany...	Electrician (London), 1897, Vol. 38, page 716..........	10 to 16	15,000	1,225,000	81.70	c
Paderno, Italy..........	The Engineer, 1902, Vol. 39, page 64..	90	13,000	....	120.00	b
Champ, France..........	Eng. Record, 1905, Vol. 52, page 648...	104	6,750	1,000,000	148.00	d
Dep't. de l'Isére, France	Elec. Review (London), 1898, Vol. 43, page 475..........	330	4,000	136,000	34.00	b
Dep't. de Jura, France.		6.5	300	45,000	150.00	d
Upper Savoy, France...		450	11,000	182,000	165.50	c and m
Chèdde, France........	Die Ausnutzung der Wasserkräfte, page 198—E. Mattern..........	455	10,000	....	30.00 42.50	a c and n
Chèvres, Switzerland...		14 to 27	9,600	1,044,000	109.00	b
Kubel, Switzerland.....		296	5,000	1,074,000	215.00	d and o
Schaffhausen, Germany.		13.8 to 15.8 11.5 to 14.8	2,700	365,000	135.00	d and p
Gersthofen, Germany..		32.8 to 34.4	6,000	812,500	135.00	b
Augsburg, Germany....		....	9,100	1,875,000	206.00	d
Heimbach, Germany...		230 to 360	16,500	2,125,000	130.00	d and q
Lyon, France..........		33 to 40	22,750	6,500,000	287.50	d and r
Mühlhausen, Germany.		24 to 30	23,000	3,075,000	132.50	b

Notes in tables 42 and 43.

a = The cost of water power development, not including dam.
b = The cost of water power development, including dam.
c = The cost of complete water power development, including electric station equipment.
d = The cost of complete water power development, including electric station equipment and transmission lines.
e = Mostly 12-hour H. P. distributed to adjacent mills at the generated voltage.
f = Severe climatic and river conditions during construction.
g = Very favorable location; cheap timber dam; transmission line only 5 miles long.
h = Includes extra real estate investment.
i = Expensive canals in rock, and very extensive concrete construction.
j = Factory installation.
k = Pelton wheels and 1,500 feet wood-stave pipe line.
l = Four interconnected plants; including also steam auxiliary.
m = Not including 5,000 H. P. necessary steam auxiliary.
n = Not including dam.
o = With 1,000 H. P. steam auxiliary.
p = Two interconnected plants.
q = 15 mile transmission line.
r = 12 mile feeder canal.

error in the estimate of cost caused by large and unexpected contingencies in construction may mean a commercial failure of the enterprise. The same consideration should be given to the estimate of contingent expenses, depreciation and operating expenses, and each other factor on which the financial life of the plant depends.

In every operating plant there is in the course of time a certain deterioration or reduction in value due to ordinary operation and the effect of the elements. In the consideration of any power plant as an investment, allowance must be made in the annual charges for a sum sufficient to keep the original investment intact. In order to accomplish this an allowance should be made on each feature of the plant for the annual reduction in value or deterioration. The amount of depreciation will vary with the character and use of the machinery or structure and should be estimated with the best possible knowledge of the conditions under which the plant will be operated, fully in mind. Such estimates should be sufficiently large to fully cover this item in order that the feasibility of the project may be correctly estimated.

The allowance for depreciation in an operating plant should be placed in a sinking fund which should be used to replace the various portions of the plant at the expiration of their useful life.

313. Annual Cost of Developed Power.—As already pointed out the annual cost of operating a plant includes:

a. Administration and operating expense.

b. Maintenance and repairs.

c. Depreciation.

d. Interest, insurance and taxes.

Each of these items will vary with the duration and the conditions under which the power plant is installed and operated. The method of estimating these charges is shown in the following estimates of the cost of operation of the Chicago Sanitary District Hydro-Electric Plant (see Electric World, Feb. 28, 1906).

Total cost of development and transmission		$3,500,000.00

ESTIMATE OF COST

Interest on investment at 4 per cent	$140,000.00	
Taxes on real estate buildings, etc	7,260.00	
Depreciation on buildings at 1 per cent	3,650.00	
Depreciation on water wheels at 2 per cent	2,027.32	
Depreciation on generators at 2 per cent	1,824.60	
Depreciation on pole line at 3 per cent	2,020.50	
Depreciation on other electrical appliances at 3 per ct.	3,995.52	
Total fixed charges		$161,137.94

OPERATING EXPENSES

Power and sub-station labor	63,240.00	
Repairs to machinery and buildings	3,700.00	
Incidental expenses	1,200.00	
Operating Lawrence avenue pumping station	43,960.00	
Operating 39th avenue pumping station	120,380.00	
Interest on investment 39th avenue pumping station	15,599.76	
		248,079.76
Total cost to sanitary district		$409,217.70
Capacity 15,500 H. P. Cost per H. P. per annum		$26.40

An interesting comparison of the estimated yearly cost of various hydro-electric generating plants is given in the various reports of the Ontario Hydro-Electric Power Commission which are reproduced in Table 44.

TABLE 44.

Estimated Yearly Operating Expenses of Generating Plant From Reports of Ontario Hydro-Electric Power Commission.

Location of Plant.	Horse power.	Net H. P. transformed for transmission.	Operating expenses including administration.	Maintenance and repairs.	Depreciation.	Interest at 4 per cent.	Water rental.	Yearly charge.	Yearly cost of transformed 24-hour power.
(1) Niagara plant	50.000	48.750	$57,900	$115,700	$ 86,800	$231,400	$52,000	$544,300	$11.16
	75.000	73.125	70.200	140,400	105,300	280,800	65,000	661,700	9.05
	100,000	97,500	86.300	172,600	129,500	345,200	77,500	811,100	8.32
(2) Middle Falls	5.200	4.990	11,875	9,500	9,500	19,000	......	49,875	10.00
Healey's Falls	8,000	7.680	16.875	13,500	13,500	27,000	......	70,875	9.10
Two above combined	13,200	12,670	23,000	23,000	23,000	46,000	......	115,000	9.08
(3) Maitland River	1,600	......	5,665	2,754	2,755	13,000	......	24,174	......
Saugeen River	1,333	......	4.840	3,247	3,247	9.984	......	21,318	......
South River	750	......	4,100	2,620	2,620	4,534	......	13,874	......
Severn River (Big Chute)	4,000	......	17,483	8,571	8,571	14,000	......	48,575	......
Severn and Beaver Rivers combined	6.267	......	23,713	13,968	14,000	25,640	......	77,289	......
(4) St. Lawrence River	......	1,200	6,864	5,119	5,118	7,151	......	24,252	......
Mississippi River High Falls	......	2.400	9,391	3,840	3,841	7,777	......	24,849	......
Mississippi River High Falls	......	1.100	6.390	2,491	2,491	4,908	......	16,280	......
Montreal River Fountain Falls	......	2,400	9.850	3.903	*21,622	8,539	......	43,914	.....
(5) Dog Lake	......	13.675	13.760	16.427	15,927	35,278	......	79,392	......
	......	6.840	11.296	10.632	10.132	24,787	......	56.847	......
Cameron Rapids	......	16.350	16.375	17.327	16,727	32,561	......	82,990	......
	......	8.250	14.390	11.478	10.978	24.008	......	60.854	......
Slate Falls	......	3.686	6.000	6.634	6.334	14.303	......	33,272	......
	......	1.843	6.000	3.868	3.669	10,400	......	23.957	......

* Including 10-year sinking fund.

314. Cost of Distribution.—Having estimated the annual cost of the development of power at the plant, the cost of distributing the power to the customer must also be considered. In many power plants the power is generated at or near the point where it is to be used and the transmission losses and costs will include its transmission through shafting, cables, and belts, or by electrical means, to the machine or appliances in which it is to be utilized. In other cases the power has to be transmitted for miles by high voltage electric currents. The units of power for which the power company will receive compensation may or may not include these various transmission losses. Where the power is distributed to a factory, the losses in transmission through shafting, belting, etc., is usually at the consumer's expense; but the transmission loss in long distance lines is ordinarily assumed by the power company and must be taken into account in the determination of the cost of furnishing power to the consumer. The losses in any system of distribution are a considerable element of the cost of the delivered power and must be carefully estimated (Section 20, page 31, et seq).

The losses in the distribution of power in various mills, factories, etc., as determined by Prof. C. H. Benjamin, are given in Table 45, page 676. The reports of The Ontario Hydro-Electric Power Commission, to which references have already been made, furnish numerous clear analyses of the cost of electrical distribution. Table 46, page 677, shows such an estimate for the delivery of power from a proposed Niagara plant to a proposed sub-station at Hamilton, Ontario. Table 47, page 678, shows the estimate of the commission on the cost of distributing power from a sub-station to an individual consumer not within the local distribution. The variations in the cost of power from the generating plant to the consumer is also well shown by Table 48, page 678, taken from the same source.

To make the delivered current available for power, a motor must be installed. This is commonly furnished by the consumer. Table 49, page 679, shows the estimated cost of induction motor service per horse power per year.

315. Effect of Partial Load on Cost of Power.—The maximum amount of work that any plant can accomplish will be done only when the plant works to its full capacity for twenty-four hours per day. Thus, if a plant has a capacity of 1,000 H. P. and is operated continuously during the twenty-four hours, the total output will be 24,000 H. P. hours of work. Under such conditions the plant can be built

TABLE 45. *Data and Results of Factory Friction Tests (see page 675).*

Nature of Work.	Total length line shafting, feet.	Diameter line shaft, inches.	Revolutions per minute.	Number of bearings.	Number of counters.	Number of machines.	Total horse power.	Percentage to drive shafting and engine friction.	Horse power per 100 sq. ft. of shafting per minute.	Horse power per bearing.	Horse power per counter.	Horse power per belt.
Wire drawing and polishing......	1,130	2½ 2¾ / 4 6	170	115	69	...	400.	39.2	.10	1.37	2.28	1.76
Steel stamping and polishing.....	580	3 3½	200	68	27	18	74.	77.	.059	.84	2.11	2.40
Average	.....	....	....	...	...	...	......	....	.08	1.10	2.19	2.08
Boiler and machine work........	530	2½ 3	150	46	47	43	38.6	65.6	.04	.550	.538	.477
Bridge machinery................	1,460	2½ 3 / 4	110	142	79	69	59.2	80.7	.04	.337	.606	.521
Heavy machine work.............	1,120	3	190	110	96	68	112.	57.	.038	.581	.665	.453
Heavy machine work.............	1,065	2 3 / 4	180 150 / 150	114	152	123	168.	54.2	.06	.799	.600	.475
Average	.....	....	....	...	...	...	......	62.3	.044	.567	.602	.481
Light machine work..............	748	1½ 1¾ / 2 3	135 135 / 135 150	101	133	250	40.4	51.2	.034	.204	.155	.095
Manufacture of small tools.......	500	2 3	114	58	314	313	74.3	53.8	.09	.689	.127	.119
Manufacture of small tools.......	990	2½ 1¾	175 136	102	202	258	47.2	51.8	.03	.240	.121	.113
Sewing machines and bicycles....	2,490	2 6	150	274	403	454	190.	56.9	.05	.397	.269	.208
Sewing machines.................	1,472	2 3 / 4	160 160 / 125	184	435	179	107.	69.7	.034	.406	.172	.154
Screw machines and screws......	1,800	2 2¼ / 2½ 3	180	180	392	428	241.	47.3	.05	.633	.291	.235
Average	.....	....	....	...	...	...	......	55.1	.048	.428	.189	.154
Steel wood-screws................	674	1½ 2 / 3	175–160 / 175	96	89	392	117.	14.5	.02	.178	.191	.130
Manufacture of steel nails........	988	2½	200	74	175	184	91.6	49.9	.035	.615	.260	.244
Planing mill.....................	165	3	267	19	40	53	39.2	73.	.08	1.52	.715	.636
Light machine work..............	275	2	175	37	27	30	8.28	48.6	.015	.109	.749	.084

TABLE 46.

Showing Investments, Annual Charges, and Cost of Low Tension Power at Sub-station. Sub-station Included (see page 675).

	Full load.	¾ load.	½ load.
Total horse power distributed...........	16,000	12,000	8,000
Total investment, including step-down stations and interswitching...........	$450,879.00	$404,879.00	$358,379.00
Investment per H. P. delivered.........	28.18	33.73	44.80
Total annual repairs, depreciation, patrolling and operation................	22,496.00	19,092.00	15,651.00
Administration, 10 per cent. of repairs, etc.	2,250.00	1,909.00	1,565.00
Annual interest, 4 per cent. of invesment	18,035.00	16,195.00	14,335.00
Total annual charges.............	$42,781.00	$37,196.00	$31,551.00
Cost of 24-hour power, including line and step-down sub-station losses...........	12.69	12.49	12.35
Cost of transmitting and transforming..	2.67	3.10	3.94
Total cost of power...............	$15.36	$15.59	$16.29

The above costs of power are based on an assumed rate of $12 per 24-hour horse power per annum for high-tension power at Niagara Falls.

at a minimum expense per unit of output and the cost of operation, fixed charges, interest, etc., will be less per unit of work done than under any other condition of operation.

For example: If a plant of 1,000 H. P. be installed at a cost of $100,000, the annual cost of operation, including fixed charges and all other legitimate expenses, may be estimated as follows:

Interest on $100,000 at 6 per cent..........................	$ 6,000
Repairs and depreciation....................................	5,300
Operating expenses..	10,000
Miscellaneous and contingent expenses.......................	4,250
Total annual cost of power..................................	$25,550

On the above basis the annual cost for each horse power of maximum load will be $25.55. If the plant works at its maximum capacity for twenty-four hours per day, the cost per horse power hour will be .292 cents. If, however, the plant is operated to its full capacity for twelve hours per day only, the total cost of power may be reduced to say $23,000 per annum. In this case the cost per horse power of maximum load will be reduced to $23 per year, but the cost per horse power hour of energy generated will be increased to .526 cents. In many cases the plant will be used for ten hours per day and for six

TABLE 47.

Showing Cost of Distribution From Municipal Sub-station to an Individual Consumer, not Covered by Local Distribution (see page 675).

Distance in miles from municipal sub-station.	Cost per Horse-Power per Annum for the Delivery of Various Amounts of Power.							
	50 H. P.	75 H. P.	100 H. P.	150 H. P.	200 H. P.	250 H. P.	300 H. P.	
2.............	$ 5.58	$ 4.20	$ 3.53	$ 2.92	$2.74	$2.60	$2.51	2200 volts.
3.............	6.89	5.20	4.41	3.60	3.25	3.10	3.03	
4.............	7.92	6.18	5.20	4.27	3.93	3.72	3.86	
5.............	8.87	7.18	5.98	4.96	4.55	4.32	4.17	
6.............	10.20	8.24	6.77	5.38	5.13	4.60	4.43	
8.............	14.10	10.14	8.40	6.97	6.24	5.79	5.58	11000 volts.
10.............	16.12	12.13	9.54	8.31	7.68	6.96	6.17	
12.............	18.76	14.03	11.12	10.12	8.42	7.96	7.22	16500 volts.
15.............	22.74	17.08	13.48	10.89	9.35	8.84	8.32	

days per week. Its maximum capacity may be utilized only occasionally, and the demand for power will vary greatly from hour to hour resulting in a load factor of perhaps fifty per cent. or less. In this case the annual cost per maximum horse power will still not exceed $23 per year, but the annual cost of average ten hour power will be $46, and the cost per horse power hour of useful work will be increased approximately to one and five-tenths cents. The cost of each unit of power under the last condition is over five times as great as in the first case mentioned, and about three times as great as in the sec-

TABLE 48.

Increase in Annual Cost From Station to Consumer Due to Investment and Losses (see page 675).

Amount of Power Delivered.	Cost of 24-Hour Power per H. P. per Annum.		
	At Niagara Falls including line and step-down sub station losses.	At sub-station.	At customer.
Full load, 2,000 H. P.	$18.54	$21.89	$26.03
¾ load, 1,500 H. P.	13.18	23.54	29.06
½ load, 1,000 H. P.	12.85	27.21	34.48

ond case discussed. It is therefore obvious that unless the conditions of use are carefully studied and conservatively estimated, they may lead to unfortunate investments and financial losses.

316. Further Considerations of Cost of Power.—The cost per horse power as previously discussed is more or less indefinite as the real value of a plant depends upon the relation of the cost of its actual power output to the returns which can be obtained from the same. These returns are dependent not only on the horse power capacity of the plant, but on the river flow and load factor which will obtain from the actual or prospective market. The questions involved are:

TABLE 49.

Capital Cost and Annual Charges on Motor Installations (see page 675).

Polyphase 25-Cycle Induction Motors.

Capacity H. P.	Capital cost per H. P. installed.	Annual Charges.			
		Interest 5 per cent.	Deprecia-tion and repairs, 6 per cent.	Oil, care and operation.	Total per H. P. per annum.
5	$41.00	$2.05	$2.46	$4.00	$8.51
10	39.00	1.95	2.34	3.00	7.29
15	35.00	1.75	2.10	2.50	6.35
25	28.00	1.40	1.88	2.00	5.28
35	25.00	1.25	1.50	1.75	4.50
50	24.00	1.20	1.44	1.50	4.14
75	21.00	1.05	1.26	1.25	3.56
100	20.00	1.00	1.20	1.00	3.20
150	17.00	.85	1.02	.80	2.67
200	16.00	.80	.96	.70	2.46

1. Continuous and intermittent horse power of stream.
2. Actual output of power which can be sold.
3. Load factor or distribution of power which will obtain.
4. Power equipment which must be installed to maintain load and give proper reserve.
5. Auxiliary power necessary to maintain service.
6. Cost of installation.
7. Cost of operation.
8. Annual expenses.
9. Actual cost per unit of power delivered.
10. Actual income per unit of power delivered.

In Table 50, page 681, have been tabulated estimates of the financial relations of various hydro-electric projects in the upper Mississippi valley. This table is based on either the actual cost of various plants constructed and in operation, or on estimates of proposed plants on which an investigation has been made. The table contains in general:

1. The approximate continuous power which would be developed at the site (in some cases with storage).

2. The horse power of turbines actually installed or proposed under the conditions of the proposed market.

3. The head which is or can be developed.

4. The net cost of the plant, *not including value for water power rights or privileges,*—i. e. the actual or closely estimated construction costs.

5. The installation costs per continuous horse power.

6. The installation costs per horse power installed, or proposed.

7. The estimated output of the plant, in kilowatt hours, which can practically be delivered to customers per year with the plant fully loaded.

8. The annual expenses, including fixed charges, operating expenses, depreciation, repairs, taxes and insurance.

9. The cost per kilowatt hour delivered at the customer's switchboard, on the basis of the sale of fifty per cent., seventy-five per cent. or 100 per cent. of the estimated practicable output.

These plants are believed to be fairly representative of the possibilities of water powers in the upper Mississippi valley.

The estimate of cost of Project No. 6 includes no transmission line as the utilization of power for a local industry was contemplated. This development promises the cheapest power of any project that the author has examined in this territory. The project, however, has not been developed and there is little possibility of its being developed for many years to come, unless it can be utilized for some local industry. The market in most of the communities within a radius of 100 or more miles is already occupied by hydro-electric or other plants which can not be replaced at any price that could be made from this development. The location is so far from main lines of transportation that the cheap power possible offers little incentive for the development of a local industry.

Plant No. 13 is constructed but, unfortunately, is delivering power at a market where coal cost is a minimum, and although this plant has been in operation for several years and is closely associated with the

TABLE 50.

Estimate of Financial Relations of Various Hydro-Electric Projects in Upper Mississippi Valley (see page 680).

Project.	Continuous H. P.	Installation H. P.	Head.	Cost *b*.	Installation Cost		Estimated practicable output delivered to customer per year.	Annual Expense *c*.	Cost Per K. W. H. Delivered.			Remarks.
					Continuous H. P.	Installated H.P.			*–50%	†–75%	‡100%	
							K. W. H.		Cents	Cents	Cents	
1	2,800	8,500	39 29 20	$1,520,000	$543.00	$179.00	16,650,000	$175,500	2.10*c*	1.40*c*	1.05	Cost does not include reservoir or flowage.
2	1,500	3,900	61	1,008,000	674.00	259.00	8,500,000	83,000	2.04	1.36	1.02	
3	1,250	3,100	50	869,000	695.00	280.00	6,970,000	72,475	2.18	1.45	1.09	
4	2,650	6,000	108	1,520,000	574.00	253.00	15,070,000	122,175	1.694	1.129	.847	
5	625	1,500	17	275,000	440.00	183.00	4,160,000	33,600	1.614	1.076	.807	
6	8,500	12,000	80	1,380,000	162.00	115.00	45,700,000	111,600	.49	.327	.245	No transmission line, power to be used at site.
7	4,300	9,000	40	2,042,800	473.00	227.00	23,300,000	163,700	1.40	.932	.70	
8	1,215	2,700	24	247,780	204.00	92.00	5,570,000	25,822	.93	.62	.465	
9	740	2,000	60	139,000	188.00	70.00	3,500,000	18,120	1.04	.693	.52	No flowage.
10	2,200	5,600	100	271,000	123.00	48.00	10,000,000	34,680	.692	.461	.346	No flowage.
11	4,000	10,000	17	2,100,000	525.00	210.00	32,000,000	150,000	.938	.625	.469	
12	6,000	15,000	25	3,000,000	500.00	200.00	60,000,000	225,000	.750	.50	.375	
13	3,750	7,500	85	1,300,000	350.00	175.00	18,000,000	110,000	1.222	.815	.611	
14	3,100	5,860	17	1,436,100	463.00	245.00	16,650,000	138,000	1.66	1.107	.83	
15	1,200	4,000	110	271,400	226.00	68.00	6,700,000	30,130	.90	.60	.45	Does not include flowage or financing cost.
16	.....	8,000	90	3,000,000	....	375.00	16,000,000	248,000	3.16	2.107	1.58	With some steam.
17	.....	8,000	90	3,662,000	....	457.00	40,000,000	446,000	2.23	1.487	1.115	With steam plant.
18	.....	3,000*a*	30	361,000	....	120.00*a*	6,300,000	47,450	1.65	1.10	.825	
19	3,250	4,500	42	750,000	230.00	166.00	15,000,000	90,000	1.20	.80	.60	With small steam plant.

a = Approximate.
b = Inc. no value for Water Power Rights.
c = Fixed charges. Operating Exp. Depreciation and Repairs. Taxes and Ins.

* = Based on Sale of 50% possible Output.
† = Based on Sale of 75% possible Output.
‡ = Based on Sale of Entire possible Output.

actual retailing of power, it has so far proved unprofitable as it is only half loaded and cannot deliver current under such conditions at as low a price as such current can be generated by steam.

Plant No. 16 is constructed and in operation. The project was extravagantly financed and has met with a number of reverses. The present cost of energy from this plant, even when fully loaded, is practically equal to the cost of energy generated from steam at the point where this power is marketed. If this plant could be combined with public utilities, using the already installed steam plants as its steam auxiliary and adding the cost of the same to its installation charges, the conditions would be as shown in Project No. 17, and if the total power could be sold or utilized at its full value of one and six-tenths cents per kilowatt hour, and to its full capacity, it would make a paying investment even under the unfortunate conditions of its installation.

Project No. 11 is a water power plant which has been installed for a number of years. Its bonds draw five per cent. interest, and the interest has been met promptly. Aside from this, however, the plant's net earnings have not been two per cent. per annum during the entire period of its existence. Practically all of these earnings have been used in litigation over flowage, the purchase of flowage lands not originally contemplated, or for extraordinary emergencies. Two dividends of one per cent. each represent the entire profit above bond interest up to the present time.

While water power investments are supposed to be very profitable, it may be fairly stated that so far as the upper Mississippi valley is concerned, in very few cases have the returns from such investments been sufficiently great to offer attractive returns commensurate with the actual investment and the contingencies involved; and in few cases with which the writer is familiar, have the returns to any water power company which is wholesaling power, been reasonably adequate when the risks and the amount of the investment are considered.

317. Cost of Power Generated From Other Than Water Power Sources.—It frequently becomes necessary to estimate the cost of power plants and of power developed from other than water power sources. This is necessary in order to determine the cost of auxiliary power plants and such auxiliary power as may be needed to assist a water power plant at times when the hydraulic power is deficient. It is also necessary to determine the cost of power with which the hydraulic plant may be called upon to compete.

TABLE 51.

Showing Average Power Developed and Its Cost per H. P. in 22 Steam Power Plants.

Output.		Operating expenses, per H. P.	Fixed charges, per H. P.	Total cost, H. P. per annum.	Cost per H. P. hr., cts.
Average H.P. developed.	No. of days per annum.				
12.4	361	$147.93	$25.40	$173.33	5.648
20.9	365	123.12	28.42	151.54	1.868
21.5	361	90.47	17.80	108.27	2.918
32.9	330	22.56	5.83	28.39	.832
36.7	365	137.25	96.70	233.95	2.811
42.4	365	86.38	63.20	149.58	1.708
53.	309	56.94	19.51	76.45	1.596
58.8	365	97.30	33.82	131.12	1.613
70.4	365	101.69	20.78	122.45	1.641
129.3	365	30.14	9.41	39.55	.871
166.7	313	15.19	4.47	19.66	.639
173.	313	22.66	5.83	28.39	3.333
210.9	290	40.33	7.86	48.19	.693
296.7	297	45.56	7.81	53.37	.749
926.	307	11.73	8.77	20.50	.691
1,010.8	306	15.70	7.74	23.44	.794
1,174.8	306	10.19	5.50	15.69	.531
1,278.7	293	10.49	6.23	16.72	.590
1,345.5	365	23.28	9.42	32.70	.820
1,352.	365	33.03	29.41	62.44	.713
1,909.7	306	13.40	6.63	20.03	.677
2,422.	306	15.67	6.73	22.40	.757

For a correct estimate of such cost, it is necessary to determine the efficiency of the various parts of the plant (see Chapter II) under all conditions of operation in order to correctly determine the actual cost of power due to the conditions of operation. The conditions for maximum efficiencies are seldom met in actual operation, and the cost of generating power is increased by the irregularities of operating conditions. In all power plants the effect of partial or irregular load affects the cost of power in the same manner as previously described in Section 315.

By far the largest amount of power generated is from fuel and by steam plants. The cost of the development of steam power is modified by the cost and character of coal used; the size and character of the machinery operated; the character of the load (that is, the load factor); the number of hours during which the plant is used per year; and the skill and ability of the engineer and fireman who have charge of the plant. Observations of the actual cost of developing power

must therefore form the basis of any accurate estimate of the cost of power production.

Mr. H. A. Foster * made actual tests of twenty-two different power plants, including manufacturing establishments, electric light stations, pumping plants, etc., and determined for each plant the power consumption per annum and its cost, including not only running expenses

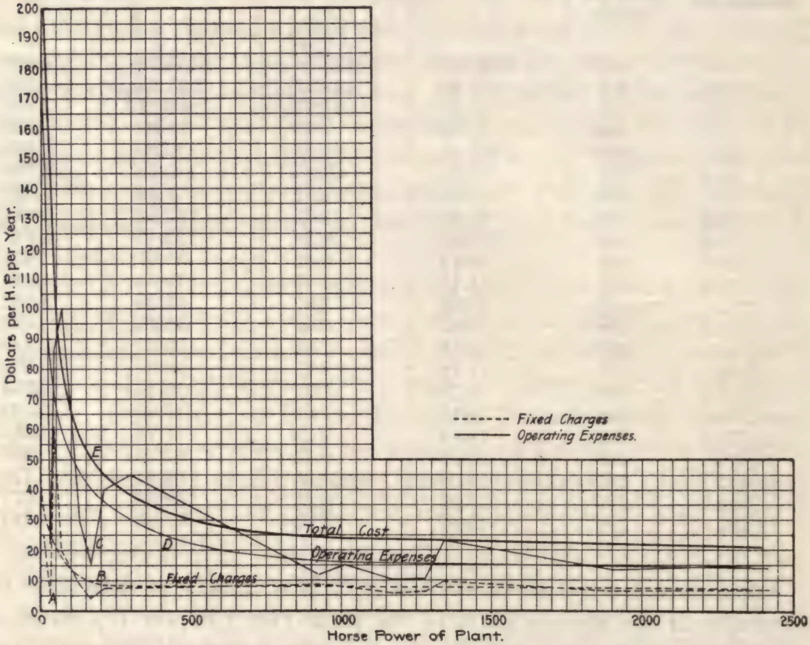

Fig. 426.—Cost of Steam Power per Horse Power per Annum in Various Plants.

but fixed charges. The cost per horse power per annum varied from a minimum of \$15.69 to a maximum of \$233.95. A summation of the results of these observations is shown in Table 51 and the plotted results of the table are shown in Fig. 426.

Mr. R. W. Conant † determined the operating expenses of various street railway power stations and compiled a table (see Table 52, page 685) which gives important information bearing on this question.

An important discussion of the effect of the load factor on the cost of power was recently made by Mr. E. M. Archibald.‡ This discus-

* See Trans. Am. Inst. E. E., Vol. 14, p. 385.

† See Engineering News, Vol. 40, p. 181.

‡ See Electrical Age, Nov., 1906.

TABLE 52. *Cost of Operation of Various Street Railway Power Stations (see page 684).*

Stations.	Capacity in 1,000 K. W.	Type.					Period averaged, days.	Load factor, per cent.	Labor.						Fuel.				General expenses per K. W. hour.	Total operating expenses per K. W. hr.	Fixed charges.
		Engines.	Generators.	Belted or D. C.[1]	Simple or compound.	Non-condensing or condensing.			No. of shifts.	Length of shifts, hours.	Total shifts, hours.	No. of men per 1,000 K. W.	Rate of pay per hour.	Per K. W. hour output.	Lbs. per K. W. hour output.	Price per net ton.	Anthracite or bituminous.	Cost per K. W. hour.			
														Cts.				Cts.	Cts.	Cts.	Cts.
Standard	3.6	3	3	D. C.	C.	C.	365	33½	3	8	8,760	1.94	0.27	0.157	2.2	$3.00	B.	0.33	0.093	0.58	0.405
10........	0.4	2	2	D. C.	S.	N.	1	24	...	....	.....	7.5	.18	......	....	2.00	.5 A.	.....	......	1.59	......
11........	1.4	5	8	B.	S.	N.	365	23	2	12	8,760	3.7	.20	.32	6.7	2.24	A.	.75	.33	1.40	......
12........	1.0	3	7	B.	C.	C.	365	23	2	12	8,760	3.7	.20	.31	4.3	2.93	B.	.63	.11	1.05	.44
13........	2.6	4	4	D. C.	S.	N.	365	42	2	12	8,760	3.0	.22	.16	6.5	1.44	B.	.47	.09	.72	.24
14........	1.6	4	4	Both	S. C.	N. C.	365	19	2	12	8,760	1.87	.15	......	....	.....	....	.....	......	.....	......
15........	1.6	10	16	B.	S. C.	N. C.	183	41	2	12	4,380	6.3	.16	.25	6.5	2.10	B.	.68	.14	1.07	.24
16........	1.6	8	8	B.	C.	C.	365	33	3	8	8,760	3.7	.20	.23	5.0	3.30	B.	.83	.09	1.15	......
17........	0.8	2	4	B.	C.	C.	365	24	2	12	8,760	3.7	.18	.28	5.0	1.43	B.	.36	.06	.70	......
18........	0.4	2	3	B.	S.	N.	365	.....	2	12	8,760	3.7	.12	......	....	.75	B.	.....	......	.....	......
19........	3.0	6	6	D. C.	......	......	31	.....	2	12	744	3.0	.17	......	....	.60	....	.....	......	.68	......
23........	0.6	3	3	B.	C.	C.	183	16	2	12	4,380	3.3	.25	.51	6.0	3.44	B.	1.03	.16	1.70	1.00
25........	4.0	10	19	B.	S.	N.	365	33	2	12	8,760	.....	.....	......	5.6	1.05	....	.29	......	.56	......
26........	0.8	7	12	B.	S.	N.	365	19	2	12	8,760	.....	.....	......	6.8	1.15	....	.39	......	1.20	......
28........	1.4	4	8	B.	S.	C.	31	23	2	12	744	5.0	.17	.37	7.0	1.63	....	.57	.08	1.02	......
29........	1.9	5	5	B.	C.	C.	365	27	2	10	7,300	3.2	.21	.25	3.9	2.80	....	.55	.23	1.03	......
30........	1.7	5	5	B.	C.	C.	365	23	2	10	7,300	3.0	.21	.27	3.3	3.00	B.	.49	.15	.91	......
31........	0.9	3	3	B.	C.	C.	365	32	2	10	7,300	4.5	.21	.29	4.4	2.84	B.	.62	.19	1.10	......
32........	5.3	5	30	B.	C.	C.	365	31	2	12	8,760	3.5	.20	.23	3.5	3.00	B.	.52	.09	.84	......
33........	1.5	4	12	Both	C.	C.	365	16	2	12	8,760	5.0	.20	.62	4.3	3.08	B.	.66	.22	1.50	......
34........	2.0	4	4	D. C.	C.	C.	183	57	2	12	4,392	5.8	.20	.18	5.3	1.00	B.	.27	.12	.57	......
35........	1.2	1	1	D. C.	C.	C.	365	37	2	12	8,700	2.1	.26	.15	4.7	1.24	B.	.29	.04	.48	......
37........	1.5	3	11	B.	S. C.	N. C.	151	11	2	12	3,624	8.0	.16	1.10	7.3	1.75	.5 A.[2]	.63	.12	1.85	......
40........	9.2	6	6	D. C.	C.	C.	365	36	3	8	8,760	2.7	.23	.17	3.0	1.60	A.	.24	.11	.52	......
42........	1.8	3	6	B.	C.	N.	365	45	3	8	8,760	5.0	.26	.29	5.7	1.60	A.	.45	.18	.92	......
43........	5.3	5	10	Both	C.	C.	365	15	2	12	8,760	3.7	.16	.39	3.7	2.70	.6 B.[3]	.50	.14	1.03	......
44........	6.0	6	12	B.	C.	C.	365	30	3	8	8,760	3.3	.24	.26	3.5	1.60	A.	.28	.15	.69	......
45........	1.0	4	7	Both	C.	C.	30	28	2	10	600	3.0	.15	.16	4.7	1.90	B.	.45	.05	.66	......
47........	0.75	4	4	D. C.	C.	C.	183	20	2	12	4,380	8.0	.12	.41	5.1	2.12	.5 A.	.54	.21	1.16	......
48........	0.6	2	2	D. C.	C.	C.	363	35	2	10	7,300	6.7	.15	.29	3.3	2.75	B.	.45	.28	1.02	......

[1] D. C.—Direct-connected. [2] .5 anthracite and .5 bituminous. [3] .6 bituminous and .4 anthracite.

sion was accompanied by various diagrams which illustrate clearly the principles involved. Two of these diagrams are reproduced in Figs. 427 and 428, page 687. The diagrams are so complete as to need no further description. The additional diagrams and the descriptive

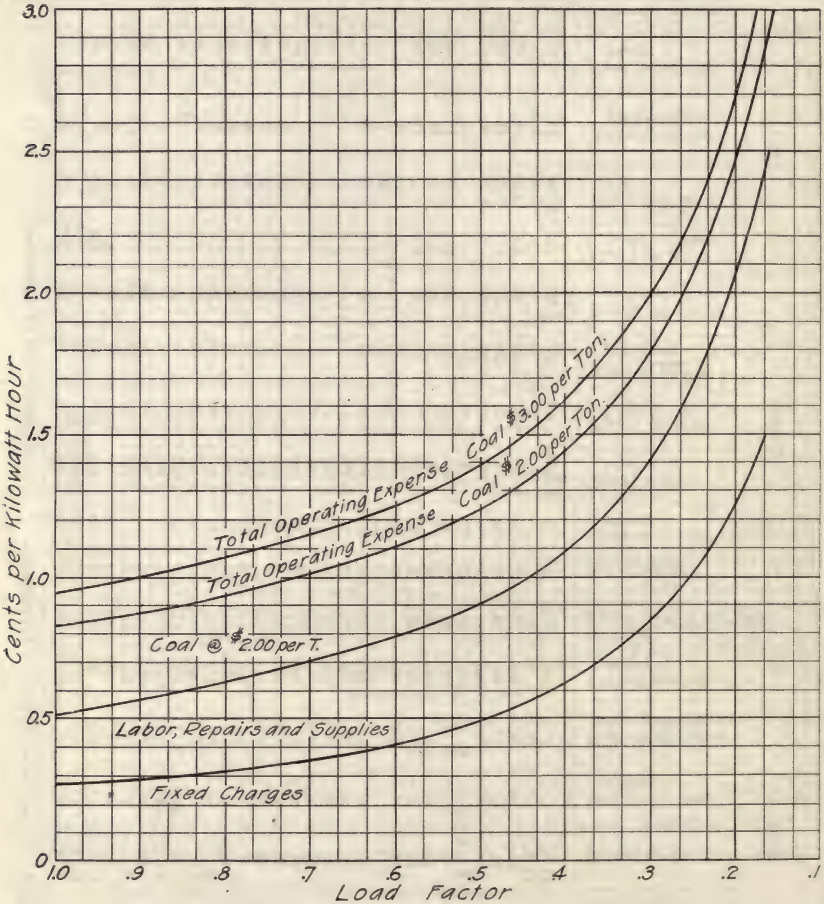

Fig. 427.—Operating Expense of a 900 K. W. Condensing Steam Plant With a 750 K. W. Peak.

matter in the paper itself may be carefully studied to advantage in this connection.

Table 53, page 688, shows the capital costs of steam power plants of various capacities and the annual cost of power per brake horse power as estimated by The Ontario Hydro-Electric Power Commission.

Similar costs for producer gas power are shown in Table 54, page 689, from the same source, and the commission's estimate of the effect on the cost of power of variations in the price of coal, is shown in Table 55, page 690.

318. Estimate of Cost of Power per Unit of Output.—In determining the cost of power in any particular case, the use of averages taken from tables and based on other conditions, is not sufficient and

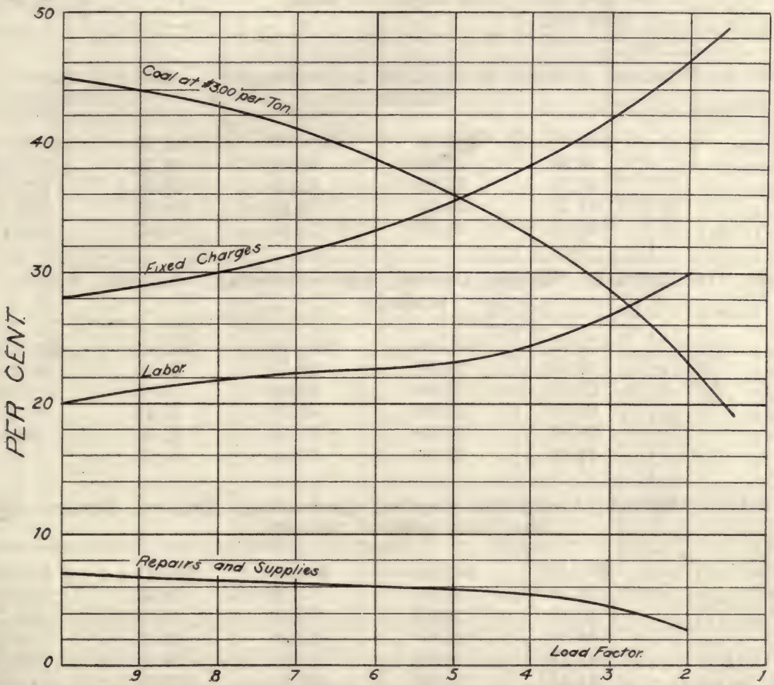

Fig. 428.—Ratio of Individual Items of Expense to Total Operating Expense of a 900 K. W. Condensing Steam Plant With a 750 K. W. Peak (see page 686).

will frequently lead to serious errors. A detailed estimate should therefore be made in which all of the factors which influence or control the cost under the given conditions are considered. As an example, an estimate is given of the cost of installation and of the power output under variations in load of a steam plant to supply auxiliary power to a hydro-electric plant during low water and low head conditions. In this case, a power station was available and no cost of building construction was involved. The total installation was 3,000 K. V. A.

TABLE 53.

Showing Capital Costs of Steam Plants Installed and Annual Costs of Power per Brake Horse Power (see page 686).

Size of Plant, H. P.	Capital Cost of Plant per H. P. Installed. Engines, boilers, etc., installed.	Buildings.	Total.	Annual cost of 10-hour power per B. H. P.	Annual cost of 24-hour power per B. H. P.
Class I.—Engines: Simple, slide-valve, non-condensing. Boilers: Return tubular.					
10..........	$66.00	$40.00	$106.00	$91.16	$180.76
20..........	56.00	37.00	93.00	76.31	151.48
30..........	48.70	35.00	83.70	66.46	131.68
40..........	44.75	33.50	78.25	59.46	117.74
50..........	43.00	31.00	74.00	53.95	106.46
Class II.—Engines: Simple, Corliss, non-condensing. Boilers: Return tubular.					
30..........	$70.70	$35.00	$105.70	$61.14	$117.70
40..........	62.85	33.50	96.35	55.50	107.10
50..........	59.00	31.00	90.00	50.70	97.73
60..........	56.00	30.00	86.70	47.42	91.34
80..........	50.00	27.50	77.50	43.86	85.41
100..........	44.60	25.00	69.60	40.55	79.19
Class III.—Engines: Compound, Corliss, condensing. Boilers: Return tubular, with reserve capacity.					
100..........	$63.40	$28.00	$91.40	$33.18	$60.05
150..........	53.70	24.00	77.70	29.83	54.63
200..........	50.10	20.00	70.10	28.14	51.72
300..........	45.90	18.00	63.90	26.27	48.83
400..........	43.55	16.00	59.55	24.84	46.12
500..........	41.25	14.00	55.25	23.73	44.21
750..........	40.50	13.00	53.50	23.56	44.02
1,000..........	39.00	12.00	51.00	23.26	43.71
Class IV.—Engines: Compound, Corliss, condensing. Boilers: Water-tube, with reserve capacity.					
300..........	$55.20	$18.00	$73.20	$25.77	$46.32
400..........	51.50	16.00	67.50	24.18	43.61
500..........	49.40	14.00	63.40	23.19	42.03
750..........	46.80	13.00	59.70	22.88	41.56
1,000..........	44.30	12.00	56.80	22.47	41.11

Note.—Annual costs include interest at 5 per cent., depreciation and repairs on plant, oil and waste, labor and fuel (coal at $4.00 per ton).

Brake horse power is the mechanical power at engine shaft.

(2,550 K. W.) in their 1,000 K. V. A. turbo-generator units. No reserve installation was necessary as the plant would only operate each year for a comparatively short time. The steam consumption was estimated at twenty-one pounds per kilowatt hour, and the boiler evaporation was estimated at seven pounds of water per pound of coal. The coal used would therefore equal three pounds per kilowatt hour.

TABLE 54.

Showing Capital Costs of Producer Gas Plants Installed and Annual Costs of Power per Brake Horse Power (see page 687).

Size of Plant, H. P.	Capital Cost of Plant Per H. P. Installed.			Annual cost of 10-hour power per B. H. P.	Annual cost of 24-hour power per B. H. P.
	Machinery, etc.	Buildings.	Total.		
10	$137.00	$40.00	$177.00	$53.48	$90.02
20	110.00	36.00	146.00	44.47	75.22
30	93.00	33.00	126.00	38.73	65.99
40	84.50	29.00	113.50	35.05	59.85
50	80.00	26.00	106.00	32.27	55.22
60	79.00	24.00	103.00	30.49	52.03
80	78.20	22.00	100.20	28.70	48.95
100	77.50	20.00	97.50	27.05	45.40
150	76.00	19.00	95.00	25.87	43.17
200	74.00	17.00	91.00	24.95	41.78
300	73.00	16.00	89.00	24.24	40.40
400	71.50	14.00	85.50	23.41	39.03
500	70.00	12.00	82.00	22.54	37.54
750	67.50	10.00	77.50	21.55	35.99
1,000	65.00	8.00	73.00	20.46	34.66

Note.—Annual costs include: Interest at 5 per cent., depreciation and repairs on plant, oil and waste, labor and fuel (Bituminous coal at $4.00 and Anthracite coal at $5.00 per ton).

Coal was estimated at $4 per ton delivered at the boilers. The detailed estimate and a graphical analysis of the same are shown in Fig. 429, page 691. From the figure it will be seen that with 100 per cent. machine factor (20,000,000 K. W. H. per annum) the cost per kilowatt hour output would be about 0.95 cents; at fifty per cent. load factor (10,000,000 K. W. H. per annum, the power output would cost about 1.21 cents per kilowatt hour; and with an output of 3,500,000 K. W. H. per annum (17.5 per cent. machine factor), the cost of pound would be two cents per kilowatt hour.

TABLE 55.

Showing the Effect on the Cost of Power of a Variation in the Price of Coal of One-half Dollar per Ton (Extra Cost per H. P. per Annum). (See page 687.)

Size of Plant.	Suction Producer Gas.		Steam.		
	10 Hour.	24 Hour.	10 Hour.		24 Hour.
10	$1.15	$2.53	Simple slide valve	$6.14	$13.47
20	1.13	2.46		5.25	11.56
30	1.10	2.40		4.71	10.35
40	1.07	2.33	Simple automatic non-condesing	3.56	7.84
50	1.04	2.29		3.37	7.41
60	1.01	2.24		3.26	7.16
80	.98	2.18		3.15	6.97
100	.96	2.12		3.12	6.87
150	.94	2.07	Compound condensing	1.75	3.85
200	.92	2.02		1.69	3.71
300	.90	1.98		1.62	3.60
400	.88	1.94		1.56	3.44
500	.86	1.89	Compound condensing water-tube boilers	1.39	3.05
750	.82	1.81		1.39	3.05
1,000	.76	1.72		1.39	3.05

319. Market Price of Water Power.—The market price of water power as further discussed (see Section 338, page 715) must be predicated on two considerations:

First: The price at which the power company can afford to furnish power and insure a fair return of its investment.

Second: The price that the consumer can afford to pay for the power. The latter amount is commonly fixed by what it will cost to produce the power by some other means.

If, in the preliminary investigation of a water power project, it is found that the cost to the power company of generating power will be greater than the price at which the power can be sold, it is, of course, evident that the plant will be a financial failure, and the scheme should be abandoned. In introducing a new source of power into any community where the power introduced will be obliged to compete with other sources, it can seldom be expected that the power to be so furnished can be sold at the same price as the power already on the market. It is at least only safe to estimate that the power must be sold at a somewhat lower figure. If the power already in use is sold or generated at a profit, a cut in price may be anticipated from the competing company; and, in the second place, as a considerable expense

is necessarily involved in the change of apparatus, etc., necessary to utilize a new source of power, consumers will be slow to make such changes unless they can do so to a considerable financial advantage.

In calculating the cost of power to a consumer, if he undertakes to generate it himself, the fair cost should be based upon interest, depreciation, operation, etc., of the plant which is necessary to be installed. If, however, the consumer has such a plant already installed, no further investment is necessary, and as the machinery installed can not be sold to advantage, the investment charges or the fixed charges on

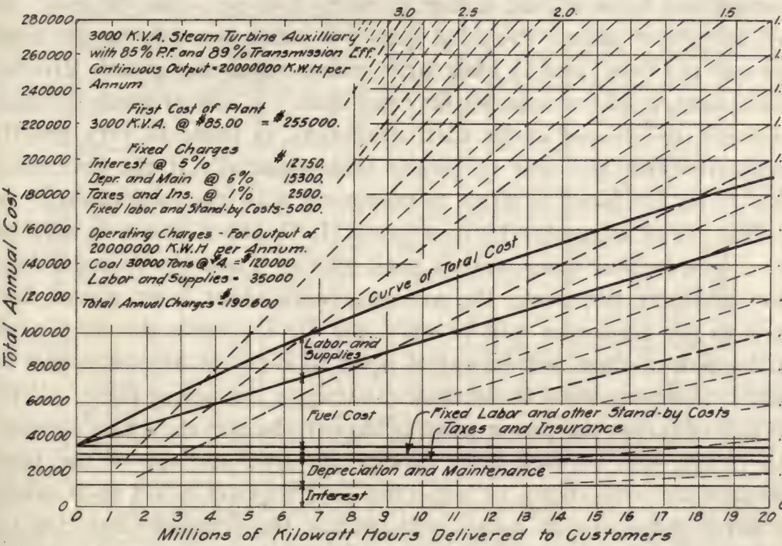

Fig. 429.—Diagram of Cost of Power (see page 689).

such a plant can not be considered, and the consumer will make a change in power only provided the power can be furnished from the new source at or below the actual cost of generation in his own plant, or at such additional cost as the convenient reliability or other desirable features of the new source of power will warrant.

In estimating the price at which the consumer can afford to purchase power, not only the price at which power is now sold but any possible decrease in the sale price due to competition or to other and more economical developments, must be considered. Better and more economical machinery in local plants, or water powers that are nearer the market and that can be developed or operated at less expense, may so

reduce the market price as to seriously affect the value of power, and hence the probability of the development.

320. Sale of Power.—Attention has already been called to the fact that if the capacity of a plant can be used for only a portion of the time, the cost of the development per unit of power, and therefore the cost per unit, used or sold, is very greatly increased. This is a matter of the greatest importance which should be kept clearly in mind in the sale of power. The load factor of many users is comparatively low. Most companies organized for the general sale of electrical power in municipalities have a load factor of thirty-five per cent. or less. In small municipal plants, the load factor is often but fifteen per cent. or twenty per cent., and in most large plants supplying greatly diversified interests, it seldom reaches fifty per cent. A sale of power to such consumers, to be used under such conditions, is liable to very greatly increase the cost per unit of power sold, especially if the maximum power to be furnished is large as compared with the total capacity of the plant. For example: If, in a 3,000 H. P. plant, power is sold on a horse power hour basis, with a peak load of 1,000 H. P. and a load factor of thirty per cent., the average twenty-four hour power furnished to the consumer will be only 300 H. P., while the total peak that the power plant will be called upon to carry at any time will be 1,000 H. P. or one-third of the total capacity of the power plant. With such sale of power the power plant is likely to be seriously handicapped. With power sold in such large blocks, the overlappings of the peak loads can not reasonably be expected to compensate for each other. The net results of such a sale will be that the company has tied up one-third of the capacity of its plant but will receive payment for only one-tenth of its capacity. It is evident that unless such conditions are realized and such a charge is made for power as will compensate the power company for the same, the power company may readily tie up its entire output and yet not receive fifty per cent. of the income that should be reasonably anticipated. If, on the other hand, the sale of power is made in small blocks, or to small consumers, it is frequently possible to greatly over-sell the total capacity of the plant and yet take care of the consumers in a satisfactory manner. That is, on account of the overlapping of the peak loads and the equalization of the load carried throughout the twenty-four hours, the total connected load sold may often considerably exceed the capacity of the plant. For example: In one water power plant, having a total capacity of about

4,000 H. P., the actual connected load is over 10,000 H. P. In many power plants the actual connected load is two or three times the plant's capacity. It is evident, however, that such a condition can exist only with comparatively small consumers, and that where a single consumer's load is a large fraction of the plant's capacity, it will not only be impossible to overload the power plant, but in addition extra ma-

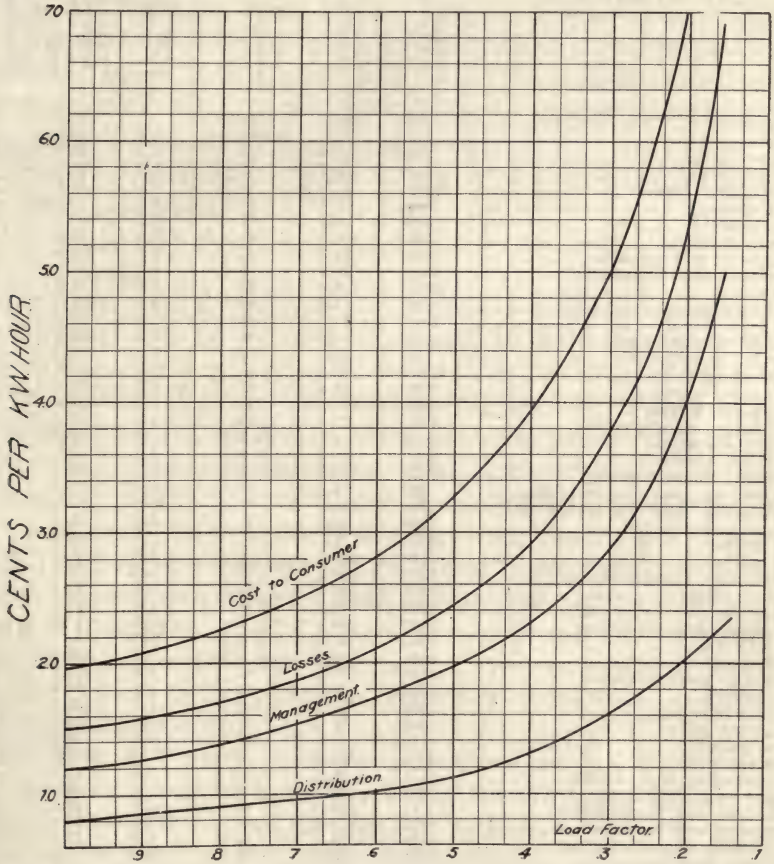

Fig. 430.—Cost of Steam-Generated Electric Power to the Consumer (see page 695).

chinery must always be installed to supply the demand should any accident happen to the regular installation.

Mr. E. W. Lloyd has compiled some valuable data concerning the

TABLE 56.

Actual Conditions Under Which Power is Furnished to Consumers From Central Stations (see page 695).

Character of Installations.	Average K. W. hours per month.	Average connected motor load, H. P.	Individual or group drive.	Average number of motors.	Percentage of average load to connected motor load.	Total number of installations.
Bakeries	1582	32.8	Group	2.7	27.8	17
Bakeries	705.3	22.5	Individual	3.1	19.5	8
Boiler shops	326.7	51.4	G	2.8	33.3	11
Boiler shops	1172	32.2	I	5.2	20.7	5
Boots and shoes	3050	39.7	G	5.8	42.8	13
Box making	1555	18.1	G	4.3	45.4	20
Blacksmiths	586	9.4	G	2.2	34.2	12
Brass finishing	5736	40.5	G	7.4	45.0	9
Butchers and packers	1990	24.8	G	2.0	36.4	13
Butchers and packers	1049	36.9	I	6.7	18.8	10
Breweries	12310	94.0	G	4.6	33.0	8
Carpet cleaning	644	14.5	G	1.6	30.1	12
Cement mixing	2009	87.5	G	1.0	24.9	4
Candy manufactory	1893	26.6	G	3.5	33.6	10
Candy manufactory	796	29.9	I	7.5	16.3	8
Cotton mills	11829	99.0	G	3.0	60.1	3
Carriage works	2091	24.8	G	3.5	35.5	22
Chemical works	4802	109.0	G	5.5	23.5	6
Clothing manufacturing	1181	23.0	G	4.0	44.5	33
Grain elevators	3842	114.4	G & I	3.8	32.6	19
Feather cleaners	2447	54.4	G & I	5.5	25.7	2
General manufacturing	6133	67.6	G & I	6.4	33.9	181
Engraving and electrotyping	863	12.4	G	2.5	46.9	8
Engraving and electrotyping	2369	46.3	I	26.7	22.5	7
Glass grinding	2760	33.5	G	3.0	36.6	6
Foundries	2057	27.7	G	2.3	43.7	15
Foundries	2419	81.1	I	7.0	21.3	18
Furniture manufacturing	1750	35.7	G	3.6	35.6	9
Flour mills	41276	148.5	G	3.1	48.1	13
Hoisting and conveying	2905	70.5	G	6.4	28.3	5
Hoisting and conveying	6562	253.0	I	20.0	13.0	9
Ice Cream	596	31.0	G & I	5.4	35.9	7
Refrigeration	4645	36.7	G & I	2.5	53.4	17
Jewelry manufacturing	2526	31.7	G	4.6	31.6	5
Laundries	676	10.8	G	2.1	34.0	19
Marble finishing	1464	19.8	G & I	1.3	51.3	2
Machine shops	4006	57.6	G	4.5	34.5	51
Newspapers	3150	47.4	G	4.8	38.0	24
Newspapers	4975	137.0	I	17.3	15.1	21
Ornamental iron works	2771	38.4	G	3.6	41.6	9
Paint manufacturing	2814	60.4	G & I	4.6	26.5	11
Printers and bookbinders	1147	20.4	G	2.6	39.5	54
Printers and bookbinders	6215	76.8	I	24.0	26.0	39
Plumbing manufacturing	3020	42.4	G	4.8	21.5	15
Rubber manufacturing	1051	26.0	G & I	15.0	24.7	2
Sheet metal manufacturing	1321	38.8	G	3.7	27.3	17
Soap manufacturing	3434	73.0	G	10.0	27.6	2
Seeds	2917	55.1	G & I	5.8	24.4	5
Structural steel	6514	176.0	I	16.1	18.5	6
Structural steel	77701	552.1	G	35.6	31.1	6
Stone cutting	7425	76.5	G & I	3.8	34.4	20
Tanners	2466	28.6	G	2.6	54.6	5
Tobacco working	3441	62.3	G	7.0	37.5	4
Wholesale groceries	2005	47.0	G & I	4.5	26.0	17
Wood working	2306	39.5	G & I	3.6	33.3	64
Woolen mills	20985	150.0	G	3.0	71.0	1
Averages	3500			6.08	33.9	951

power loads on various central states from various classes of consumers. This data is given in Table 56, page 694.

The increase in the charges for power to consumers on account of the variation in power factor is illustrated by Fig. 430, page 693, taken from the paper of Mr. Archibald to which previous reference has been made.

321. An Equitable Basis for the Sale of Power.—It is essential, in order to establish an equitable basis for the sale of power, that some additional factor besides the units of power furnished be considered in determining the basis for the prices charged. One of the most equitable bases for the sale of power is apparently:

First: A service charge to the consumer of a fixed price, based on the peak load carried.

Second: To this should be added a price for the units of power actually furnished.

The fixed price should equal the interest, depreciation, etc., on the capacity that is to be provided or set aside to carry the peak load of the customer. The unit price for power should be an equitable charge for the quantity of power which will actually be sold. Where both of these quantities are fixed, a net price per horse power per year, or a total price per annum for the power to be furnished, can, of course, be arranged equitably. The main idea in establishing a price for power is to keep clearly in mind the factors that enter into the sale of power, so that in making a contract for the use of power the rights of both power company and consumer shall be duly considered. The sale of power at a profit is one of the most essential features in the management of the power plant, and many plants have been wholly or partially financial failures on account of the ignorance of the basic principle on which power should be sold.

The method of charging for power outlined above is illustrated by the charges for electric current furnished from Niagara Falls, by the Cataract Power & Conduit Company of Buffalo, as given in the Engineering News (May 26th, 1898) as follows:

"All payments for power are to be made monthly and the amount of each monthly payment will consist of a charge for service, and in addition thereto, a charge for power. The charge for service is $1 per kilowatt per month, and this charge will depend only upon the amount of power which the user may require the Cataract Power & Conduit Company to keep available and ready for his use. The monthly charge for power will depend upon the aggregate amount used, as

determined by integrating meters installed by the Conduit Company upon the premises of the consumer. The charge for power will be determined from the following schedule:

Units (K. W. hrs.) used per month.	*Charge per unit* For current up to	*For the excess.*
Up to 1,000	1,000 units, 2.0 cts.	2.0 cts.
1.000 to 2,000	1,000 units, 2.0 cts.	1.5 cts.
2,000 to 3,000	2,000 units, 1.5 cts.	1.2 cts.
3,000 to 5,000	3,000 units, 1,2 cts.	1.0 cts.
5,000 to 10,000	5,000 units, 1.0 cts.	0.8 cts.
10,000 to 20,000	10,000 units, 0.8 cts.	0.75 cts.
20,000 to 40,000	20,000 units, 0.75 cts.	0.70 cts.
40,000 to 80,000	40,000 units, 0.70 cts.	0.66 cts.
Over 80,000	80,000 units, 0.66 cts.	0.64 cts.

322. Value of Improvements Intended to Effect Economy.—In many plants the first cost of an installation is an important matter and must sometimes have a greater effect than the interest and depreciation charge would seem to warrant. In most cases the plan should be studied in detail and improvements introduced or rejected on the basis of their true financial value. Such consideration should usually be made on the following basis:

Dr.		
Investment required to effect improvements	$............	
Interest on investment..................		$............
Depreciation on improvements............		$............
Extra expense of operation and maintenance		$............
Total annual cost of improvement...		$............
Cr.		
Saving in power (or in expense) effected by improvement	$............	
Annual value of saving effected...........		$............
Net annual gain or loss due to improvement		$............
Capitalized value of power (or expense) effected by improvement................		$............
Net capitalized loss or gain effected..		$............

323. Value of a Water Power Property.—It has frequently become necessary in this country to condemn water power privileges on account of the necessity of securing public water supplies or for other public purposes. Under such conditions it frequently becomes necessary to estimate the value of the water power property. When such matters are brought into court and various witnesses are heard on the subject, it is commonly found that very great differences of opinion exist as to the value of power. These differences of opinion are largely the result of entirely different points of view.

To those who have carefully followed the discussion of the hydrograph, and the estimate of power based thereon, the great variations that occur in the potential power of streams at various times in the season, and in the various years, are obvious.

It is apparent that different engineers, even if they take carefully into account these variations in power, may differ very greatly indeed as to the extent to which the power can be economically developed.

The question of pondage as discussed in Chapter VII also has a very important bearing on this matter. It is only by a careful study of the whole question and the consideration of the power market that even an approximately correct answer to this question can be given. The value of such a plant may be considered in a variety of ways:

First: Its value if intelligently and recently designed, may be represented by the cost of its reproduction plus a certain value for the water power rights.

Second: Its value may be computed on the capitalized net income that the plant can or does earn.

Third: The value of the plant may be considered equal to the capitalized value of the most economical plant that can be installed to furnish power at the point at which the power is to be used.

By the term "most economical" is meant not necessarily the one lowest in first cost, but the plant that, when considered in the broadest sense, will furnish power, all things considered, at a less cost than from any other source of power. The subject is a very broad one and one that needs thoughtful consideration and study.

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CHAPTER XXIII

FINANCIAL AND COMMERCIAL CONSIDERATIONS

324. **Financial Considerations.**—Every engineer who is called upon to advise his clients as to the feasibility of a proposed water power project must carefully consider not only the engineering problems involved but also all financial aspects of the project, for the entire commercial success depends on its financial feasibility. It is not enough that the power be constant and sufficient in quantity, that the plant be well designed, and that the cost of the same be reasonable; but there must also be a market in which the power can be utilized to advantage and the price at which the power can be sold in competition with all other sources of power must be sufficient to pay fixed charges and all other expenses involved in the construction and operation of the plant, and afford a fair return on the investment to those who assume the risk of the undertaking.

It is a common belief that any water power development will be profitable. As an undeveloped water power is a continual waste of energy, it is commonly assumed that the saving of this waste is bound to result in a profit to those who acquire the property and develop the power. That many water powers cannot be developed at a profit under present conditions is a fact that in many instances is learned by an owner only after large and unwarranted expenditures.

325. **The Hazards of Water Power Investments.**—The development of a water power is not a simple method of assuredly capitalizing the waste energy of streams. In the first place, the hazards involved both in the construction of such properties and in the contingencies of their operation and maintenance are considerable. With the advent of electrical transmission, coupled with the popular conception that water powers were excessively profitable, and that by means of such developments the waste energy of water could be advantageously turned into dividends, investors eagerly sought water power investment. The results have not always been satisfactory. The first large Niagara hydraulic development operated for twenty years without a dividend. The great developments at McCalls' Ferry on the Susquehanna River, at Massena on

the St. Lawrence River, and at the Sault St. Marie, are examples of financial catastrophes, the list of which could be greatly increased if further illustrations were desirable. Foreclosures and sales of water power properties have been very common. In one case, the investors in the bonds of a water power company realized less than five per cent. of their par value. In other cases, plants have been dismantled and abandoned. It is of course apparent that such projects were ill advised and should never have been undertaken; or, if attempted, undertaken on a more conservative basis. But the history of every business is full of investments of this character, and no line of business has ever been developed in which the paths of such developments have not been strewn with the wrecks of ill advised projects. A question as to the ultimate success must accompany every new enterprise and throw a doubt on the wisdom of its projectors and the advisability of the investment. This is especially true in water power development.

326. Hazards of Water Power construction.—Plans are rarely made for large and important structures that do not require more or less modification during construction. Unless this fact is duly appreciated by the designer and liberally allowed for in the estimates of cost, such estimates have always been found more or less inadequate to complete the structure. The hazards of construction increase with the difficulties of construction. In the superstructure of one of the large buildings of Chicago, designed by one of the best architects of that city, and which was completed without any serious mishaps, the cost of extras increased the original contract price almost twenty-five per cent., or about $700,000 on a $3,000,000 estimated cost

When a structure is built in and across a river and the work of construction is subject to unusual hazards which cannot be foreseen, due to conditions that cannot be fully predetermined and to the contingencies of flood, the ultimate cost cannot be accurately determined and is frequently greater than the estimate that will usually be made.

The engineer designing a plant, if unfamiliar with the contingencies of actual construction, can scarcely conceive the unforeseen circumstances which may and frequently will occur when his plans are being carried out in the field. An estimate in detail of the reasonable cost of each feature of the work can seldom cover all the costs involved in the construction of such a plant and must be over-

estimated or a large contingent estimate must be added in order to cover the actual cost. A water power project which will pay only fair returns on a close estimate of cost, is seldom worthy of serious consideration as unforeseen expenses will often make it a losing investment.

In a recent water power development which was under advisement for a number of years, and which was perhaps as thoroughly considered and as carefully planned, both in design and in methods of construction to be pursued, as any other development of late years, the cost of the finished structure exceeded the estimate by thirty-three per cent. The estimate of cost was about $21,000,000; the actual cost about $28,000,000. In another case, where estimates were carelessly made, the original estimated cost was $800,000, and the actual cost of the complete development about $2,500,000.

The unexpected extra costs of such developments, due to unforeseen delays, are often serious. The interest on bonds must be met semi-annually or annually from date of their issue; hence interest during construction is an important item in the cost of development of any industry, and is an item which is particularly uncertain in water power developments. In a recent development of this kind, a flood (the most extraordinary that had occurred on the river within the known records) not only caused a loss of approximately $40,000 to the work under construction, but was followed by continuous and unusually high water for the year following, so that not more than ninety working days were available within the year. In the same project, an ice jam in the spring carried out all the trestle and false works, involving a loss of perhaps $10,000 more. These casualties created a delay of more than a year with an extra interest cost of approximately $100,000.

Such hazards are more frequently present in all classes of hydraulic projects than in those of almost any other kind of developments. While care and experience with water power projects may perhaps finally result in greater consideration and more liberal estimates to provide for contingencies or changes which are likely to occur, still the contingencies exist, and such investment will, in the future as well as in the past, frequently be under-estimated and the actual costs of construction will require greater investments than the optimistic projectors will think possible, even when money is judiciously and cautiously expended.

It is of course obvious that a project which may be very attractive on a basis of an investment of $800,000 is likely to be a serious failure on the basis of an investment of $2,500,000; and even a project which seems attractive on the basis of a $21,000,000 investment may be seriously handicapped by an expense of $28,000,000 unless there is a prospect of extraordinary returns on the basis of the original estimated cost.

327. Contingencies of Maintenance.—Even after a plant is once constructed the contingencies are not removed. Within the last few years, a single flood caused a loss of $300,000 to one water power development. This loss resulted from an extraordinary condition which could hardly have been foreseen and which would occur but rarely under similar conditions. In this case the gates of the plant were so clogged by a rush of logs that the flood waters, unable to pass the dam, cut their way entirely around the structure, which not only caused the loss stated but put the plant out of service for more than a year (see Fig. 412, page 652). In another case, a dam was as seriously injured by a flood produced by the destruction of a reservoir dam built by a different engineer. The dam destroyed possessed sufficient strength and capacity for all contingencies of normal river flow that were liable to occur, yet the unexpected destruction of another structure, afterwards built above it, caused an extraordinary condition that resulted in a loss of perhaps $150,000 and put the plant out of commission for more than a year. In neither of these cases was the casualty produced by a fundamental defect in the design of the structure involved, yet the loss was no less serious.

Losses through defective designs are very numerous. In a western plant which was just about to be started, the whole foundation was washed out from under the dam, leaving the structure suspended from the rock walls of the narrow canon. This loss was undoubtedly due to defective design and construction, and cost the company at least $125,000 and a year's delay in beginning operation. Numerous instances could be recited where either fundamental defects or extraordinary conditions have destroyed or seriously injured dams and water power plants and have caused serious losses to their owners. Mankind is fallible; our knowledge of the possible activities of natural forces is limited; the effect of the possible combination of all of the known and unknown factors can never be

clearly seen or appreciated; yet, these contingencies are ever present and must be considered by the water power investor.

328. Elements of Cost in Water Power Development.—Seldom can a water power be developed at a first cost which compares favorably with the cost of developing a fuel-power generating plant, and if the water power company is an independent concern, which develops its source of power to "the average minimum for the maximum six months" and must at the same time maintain the market demand for power at all times, it must install an auxiliary fuel power plant to develop power at low water stages. Such auxiliary plant must frequently have a capacity almost or equally as great as that of the water power plant itself. Under such circumstances, the cost of auxiliary power development must be added to the cost of the water power development.

One of the advantages of power generated by fuel is the low cost of small installations. While the cost of power per unit developed is frequently large in such installations, such costs are usually within reasonable and possible limits of expense.

A steam plant can always be constructed of a size proportional to its prospective market and can be increased and enlarged as its market demands, to any extent and without over investment.

Small water powers of a few hundred horse power may sometimes be developed so economically, and operated so cheaply that they can compete in places where the cost of steam power is high, but seldom where steam costs are low, unless the plant is built or combined with industries actually using power.

A water power to be economically developed must practically be developed to certain proportions regardless of its market. Powers on large rivers can seldom be developed and operated successfully in a small way. On a given river it is almost as expensive to develop a small amount of power as to develop the stream to its capacity. The same dam with the same appurtenances is usually necessary, whatever the development. The same operating force will usually be required whether the plant is fully or partially loaded, and whether the development is partial or complete. If the power is transmitted, the same towers required for 5,000 K. W. will carry 10,000 K. W. successfully. The completed development will involve a larger power house, a few more turbine, generators and equipment, and a little larger transmission wire. These are usually

the only extra expenses involved. Hence, on large streams, the project must be sufficiently large to pay, and can be made to pay only when an adequate load is secured. The investment in these plants is so great that they can never be built except for an existing market which will provide at least their principal load, unless industries are developed in connection with them. Both fixed charges and operating costs begin at once when the plant is constructed, and interest starts with construction. A market must be obtained immediately on completion in order to meet fixed expenses or the plant must go into bankruptcy. In almost every case, therefore, these plants enter competition in a market already supplied with power and are under certain disadvantages when the power is placed on the market.

329. The Financing of Water Power Projects.—It is a comparatively easy matter for an operating industry that is showing fair returns on the investment to secure the amount of money necessary for reasonable expansion, or for its current business use. The financing of a corporation whose property and business are both a matter of future development, and necessarily more or less speculative, is a very different matter.

By a "speculative investment" is meant any investment dependent for its success on the development of a future productive business of any kind whether it be the manufacture of mercantile products or power, the success of which depends on the judgment of men more or less familiar with the business or expert in its successful development. The speculative elements depend upon whether or not, all conditions and things considered, the business can be developed for such a cost, maintained and operated at such an expense, and obtain and maintain a market at such prices as will make the enterprise successful to those who invest their money in the business, whether such investment be made in the shape of bonds, preferred stock or common stock.

It is seldom that a water power company can secure the necessary funds from its own stockholders to complete its project except where a market at a known price is absolutely assured and then only where the project is comparatively small and high returns are believed to be certain.

To finance a water power project where the cost of development amounts to several million dollars, is at the present day practically

impossible except through the agency of investment houses who make a business of financing such industries. The securities of such projects are taken by investment houses to be offered to investors including men in all ranks of life.

330. **Requirements of Investment Houses.**—Before a reputable investment house will undertake the financing of such enterprise and endorse it with the prestige of its name and reputation, guaranteed by the great care they have previously exercised in financing such properties, the projects must be carefully examined by experts of reputation, men who are of the highest ability and experience, who can and will vouch for the technical features of the construction, for the expense involved, for the market available, for the legality of the enterprise, and in fact, for its probable complete commercial success. Such houses will not lend their assistance to a project of this kind without fair returns both for themselves and for their clients.

331. **Securities.**—The securities issued for the development of water power properties are usually bonds, preferred stock and common stock. Bonds and preferred stock represent the actual cash investment, and common stock represents the speculative element or prospective profits. The bonds issued are nothing more or less than a subdivided mortgage; they are protected by a deed of trust made to a trust company who represents the bondholders. The issuance of bonds permits a division of the mortgage whereby any investor may purchase such portion of the mortgage as his finances and opportunities permit. Bonds are usually issued in amounts of $500 or $1,000. Bond interest, like the interest on any mortgage, must be paid when due or the property will be subject to foreclosure and sale. They run for a certain time and must then be paid and retired, although they are sometimes payable at any interest date, at a certain advanced price. Their value depends upon the success of the project. Their security is increased when they represent only a part of the actual investment and when there is an equity represented by an actual cash investment, which may be in the form of second mortgage bonds, preferred stock and common stock. The public will not be induced to invest in such bonds at only a moderate rate of interest without some form of stock bonus, that is, without a share in the prospective profits. This is almost a self-evident statement. Would any sane investor purchase a five per cent. bond on any undeveloped business when he can secure an equal return

from a farm or other real estate mortgage? If a stock bonus can be obtained with such a bond, it will give the purchaser a chance not only of increased return but of increased capital value, and such an investment at once becomes more attractive. On the other hand, a high rate of interest on bonds sufficient to induce the public to purchase such securities issued on speculative industries may be fatal to the success of the project.

Interest on bonds must be met promptly each interest day to avoid foreclosures, while stock must await surplus earnings for its dividend. On account of the necessary payment of interest on bonds, many corporations, in the face of unexpected difficulties and delays in construction and in securing a market to meet fixed charges and operating expenses, have found it impossible to meet such expenses and have ended in failures; whereas, if they could have been financed with fewer bonds or bonds at a lower rate of interest, they would have been able to have survived and ultimately to have been successful.

332. **Stock.**—Preferred stock is always a junior security to the bonds. The dividends (or interest) on preferred stock are ahead of dividends on common stock but are payable only when and as they are earned. They represent an actual investment, and sometimes share in the earnings of common stock after dividends on such common stock reach a certain amount. They can usually be paid off and retired under certain conditions. They are not a loan on the property but commonly have a certain preference in the proceeds from the property if sold and if such proceeds are more than sufficient to liquidate the bonded indebtedness.

Common stock always represents the speculative feature of an investment even when fully paid at par value. They are sometimes the only securities issued, and share in the net profits of the venture. They are always junior to preferred stocks and bonds, and represent, through their majority holders, the business management of the project.

The great advantage of stock investments to a company is due to the consequent reduction in fixed interest charges. The development of a market commonly takes a considerable period and unless a company can earn at once fixed charges and operating expenses, a considerable amount of extra capital must be provided beyond the cost of construction to carry the venture beyond this period and place it on an earning basis.

Stock investments are not popular except to the few who can come in direct touch with a project and know or believe, from such contact, in the possibilities of large returns. In a large project, therefore, very few will invest in stock only without some security representing a first lien on the actual property; for stock alone represents the maximum risk and affords too little stability to the investment.

In the case of water power developments, bonds bearing five per cent. or six per cent. interest cannot be made acceptable to the purchaser except at a discount and by a gratuitous distribution of junior securities representing the speculative side of the project. Such bonds can, however, usually be sold at a comparatively low discount if, in addition to these securities, the buyer receives a junior security which may possess a value and an earning capacity if the project is successful.

333. **Financing With Bonds Only.**—If, for example, a water power project financed on this basis, where the cost of construction and financing will be approximately $2,000,000, and if in addition to the bonds amounting to this total sum and bearing, say five per cent. interest, a speculative stock, which may be regarded as representing prospective profits, or water power rights and privileges, be also issued in the same amount, investors will often take such bonds or securities at a reasonable price from a responsible financial house of experience if they also receive as a bonus say fifty per cent. of speculative stock. Where care has been taken, the bonds are a reasonably safe investment. If the junior securities or stock should ultimately pay ten per cent., the net result to the bond purchasers would be ten per cent. on the actual total investment, which is certainly no greater return than should be earned in such a hazardous investment.

The projectors of the development, or the parent company, usually base their entire hope of reward for their endeavors in such projects on such portions of the stock as they are able to reserve from the cost of financing. They borrow the capital on their property and credit, and with the bonds representing a first lien and fixed returns, they give such proportion of the stock as the market demands.

334. **Financing With Stock and Bonds.**—If the projectors of the enterprise, or the stockholders of the parent company, make an actual investment in the junior securities or stock, thus placing the primary securities on a sounder basis by virtue of an equity in the

work, of fifty per cent. or more, the primary securities or bonds can often be sold at a low discount without a stock bonus. In such case, however, the stockholders have invested their money in a security which is secondary to the bonds issued and which involves most of the risk. Such an investment will not be made without a reasonable assurance that the returns in the same will be large. In other words, on the basis of a $2,000,000 investment, if $1,000,000 is paid from bonds and $1,000,000 from stock, under present conditions it would be practically impossible to secure investors in the original stock of the company, unless an earning of at least fifteen per cent. can be reasonably anticipated on the entire investment, in which case the larger earnings would be secured by the owners of stock who have risked their money practically without security. To effect this result, it would be necessary to issue stock to the amount of $2,000,000 and sell it on a paid-up basis of fifty cents on the dollar. The stockholder would anticipate an ultimate increase in the value of his holdings and a large return commensurate with his extra risks. A large return under such conditions must be possible, for the stockholder's property is the guarantee of the bonds which must be protected first, both for interest and principal. Here again the total earnings of the plant, if successful, would show fifteen per cent. on the actual investment necessary for its financing and construction, and unless the prospects of such earnings are fairly assured, a development along these lines is impossible. There are seldom any other methods by which these projects can be financed. The projects are speculative and appeal to only those who have speculative instincts.

In the first instance cited, the purchaser of the primary securities purchases what he believes to be a fairly safe investment, with the incentive of a certain amount of bonus stock which he hopes and believes may net him an additional return rather larger than he can secure from any other line of investment, and when such bonds are purchased from reliable investment houses who have made a specialty of certain lines of investment and who have had long and valuable experience in a given line, the purchaser of such securities is, in the long run, seldom disappointed in his receipt of some additional value besides his investment in the bonds of the company.

335. **Possibilities of Financing.**—It is believed to be entirely impossible at the present time to finance a $2,000,000 water power project

or to develop any speculative industry by the issue of $1,000,000 in bonds and $1,000,000 in stock to be sold at their par value. There are few investors who can come into such close contact with the details of such an investment as to make them feel assured that they will ultimately secure a suitable return on their stock, and investors who have not the knowledge or opportunity to assure themselves of the probable prospects of the investment must depend upon the reputation of the investment house, the projectors of the enterprise or the engineer on whose judgment the value of such securities rest, and such investors are not satisfied to take uncertain risks without prospects of large returns.

336. **Fair Returns.**—It is believed that fifteen per cent. on the total investment in a water power project is not more than sufficient returns when the risks in development, maintenance and market are considered; and that if the stockholders furnish an equity to the holders of bonds issued for only a portion of the cost, such stockholders who have taken such additional risk should be able to secure all of such returns on the total investment as are not required for the payment of bond interest.

That a prospect for the returns mentioned is necessary for the development of such enterprises is well established. Mr. Gifford Pinchot, perhaps the most ardent advocate of the government ownership of water powers, made the following statement before the National Waterways Commission in 1911:

"I should like to be understood as asserting with a good deal of vigor that I believe investors who go into water power development should be given a much more generous return on their investment than men who go into a less hazardous business, for the risks of a business of that kind are certainly very large. The public needs the development of water powers.

"I can say, for instance, that it is wholly impossible to expect general water power development under present conditions on a six per cent. basis. The risks are too large. Ten per cent. or fifteen per cent. would be more like what is required to induce capital to go into that field."

The author knows of no water power that can show any such earnings on its cost, although many were installed with such hopes. There are doubtless some powers developed by public utility companies, or by other large companies commanding markets of consid-

erable magnitude, where such returns have been made; but such results are seldom realized.

337. **State Stock and Bond Laws.**—The above methods for financing water power industries by the issue of speculative stock at less than par, and which when honestly carried out are morally unobjectionable, are impossible under the present laws of many states. Under such laws, bonds can often be sold at prices as low as seventy-five per cent., but stock, the investors in which must take practically all the risk in any reasonably speculative project, must be sold at par. Such requirements would be proposed only by those wholly unfamiliar with the necessities of financing.

If six men go into a business venture, each investing an equal share, and one, on account of his limited resources, is obliged to borrow part or all of his share, he should nevertheless be entitled to no less a return from the venture on account of his financial disability.

If an earning of fifteen per cent. on an investment is legitimate, such an earning should be no less legitimate if the projector or projectors, being without financial resources, are, through their ability, integrity and technical knowledge, able to borrow a part or all of the capital required to make such an investment.

The stock and bond laws of many of the states are, in their present form, simply a protection to the capitalist and discourage enterprises by the men of small means. Such laws are entirely unfair and are in no sense a protection to the investing public. Let bonus stock be issued, let it recite on its face just what it is, that it is speculative in character and dependent upon the success of the plant for its value. Why should not the buyers of low interest bearing bonds share in the returns of a speculative company, and why should not the projectors of a worthy enterprise who must shoulder all of the labor and risks and whose securities are secondary to the primary securities, buy their stock below par or even secure it without cost otherwise than for their enterprise, reputation and efforts, provided that the total profits secured are only those fairly warranted by the risks and hazards involved?

The attraction in any business enterprise lies in the safety of the investment, in the magnitude of the returns and in the ease and security with which the business can be developed. Investments are and can be controlled in a very limited way. The investor wavers between perfect security and possible large returns. The risk which

he will take is usually a function of his necessities and purposes, or of his individual inclinations, interests and experience. The conservative and wealthy citizen, the magnitude of whose fortune will assure a satisfactory income at low rates of interest, will be satisfied with an investment in government bonds at a rate of interest of three or four per cent. The citizen of means whose fortune thus invested would not furnish a sufficient income for his purpose, will seek investment in municipal bonds and in high grade commercial securities that will assure him with reasonable safety five or six per cent. on his capital. The individual of small means who wishes to augment his income by the earnings of such small savings as he has been able to effect, will either be contented with safe and small returns, or will seek to still further increase his income by larger returns at the expense of greater risks, and will seek investments in less assured commercial enterprises that will pay possibly seven or eight per cent. The business man familiar with certain lines of enterprise and commercial development in which he invests not only his money but his energy and credit, demands a large return which is commensurate with the risk and hazard which he knows or believes to exist; and no enterprise which will not fairly promise adequate returns commensurate with the risk involved, will receive the financial assistance necessary for its development. *This is a natural law which is beyond the reach of legislative enactments and entirely beyond public control. The congress of the United States or the state legislature may pass such laws as will seriously handicap an investment and reduce or greatly limit its value; but they cannot enforce further investment in such lines, and their restrictions may be such as to practically prevent all further investments, or to cause the use of illegitimate methods.* There is no man more eager for large returns than the man who has only a small amount of capital to invest, for only by such means can his earnings be made of material magnitude. Intelligent attempts to protect the investing public are both desirable and necessary; but many of the laws now in force hamper legitimate enterprise while they do not prevent the swindling operations of the unscrupulous promotor.

338. **The Measure of the Value of Power.**—The measure of the value of power in any community is the cost of developing such power by the means most ordinarily used or which would ordinarily be used for the development of power under the conditions, and in the qualities used or to be used, in the particular location.

In fixing the rate at which power may be sold, there are two natural limitations which must be considered. The higher limit is the price "that the traffic will bear." In other words, the highest price which can be established which will not discourage the use of power or will not be so high that large users of power will continue to generate power themselves or will be induced to undertake its generation rather than contract for its purchase. The lower limit is the actual cost of the power to the water power company. Between these two there is frequently a wide range in values.

Before the installation of the water power plant is begun, the higher price is the true value, but the moment an installation is completed, in order to secure immediate custom it becomes practically necessary to reduce the price for commercial reasons in order to induce power users to become customers.

In the installation of hydraulic plants, the upper limit is the cost at which power can be produced from fuel by the methods which would most ordinarily be employed, which criterion combines a consideration not only of economy in generation but practicability in the installation and operation of the plant, and is therefore, not represented by what an ideal plant should do under ideal conditions, but by what similar plants ordinarily accomplish under similar actual conditions of operation.

The higher rate for commercial reasons cannot obtain, as there would be no inducement whatever for individuals using any considerable quantity of power to become customers, for the price they would pay would be the actual cost of such power as generated by their own plant, and there would be no inducement offered to become customers of the water power company. The higher price is therefore a commercial impossibility, and the lower rate is evidently inequitable for there is no valid reason why a private company or the public, who has risked nothing, should have the full benefit of any commercial venture. The public, including all consumers of power generated by such a plant, will from commercial necessity receive a portion of the benefit from the use of such power, as a foregoing conclusion from the commercial conditions that follow its development. Rarely, if ever, can a water power company secure the higher return for its output. A water power company must, as a rule, supply power to a market which is partially at least already supplied with power from other sources.

Investments in power generating machinery of some kind have already been made. Fixed charges have already been incurred and the water power company finds, therefore, that it must, in order to introduce its product, sell power below the station cost of producing the same by means of steam plants, and not on the basis of fixed charges plus operating expenses of the steam plant, which has been the true measure of the value of power to the date of introduction of power from the new source. Only in cases where the market developed is entirely new and where there are no fixed charges for previous power plant installation to be met, can a power company hope to realize even a part of the fixed charges of the steam plant from the sale of water power. Even under such conditions a material reduction must be made in order to induce customers not to install isolated plants of their own for the production of such power, but to purchase the power developed from the water power plant.

339. **Difficulties in the Sale of Water Power.**—If power generated by steam is costing on an average of one and one-half cents per K. W. hour and power can be generated by water at an average of .75 cents per K. W. hour (which is a fairly accurate statement of the actual average costs), it would appear on the face of these figures that water power must indeed be profitable. A consideration of the facts will show that even on this basis, water power may prove unprofitable. In the first place, it must be remembered that not all water power plants can produce power even under the best conditions of load at the cost stated. In the second place, few plants can produce power at such figures without the sale of a large portion of their possible output on the most advantageous basis of loading. In the third place, a water power company can never secure a price for its output equivalent to the actual cost of power at the point where it markets its product, except when the development is undertaken by a company actually utilizing power in its business, and with a business large enough to utilize practically the entire output of the hydraulic development. For example, consider a community of thirty or forty thousand population in which a well planned steam electric plant, working under the conditions which ordinarily obtain in such communities, can develop power at its switchboard for about 1.6 cents per K. W. hour: of this amount, approximately .5 cent per K. W. hour is fixed charges, and 1.1 cents per K. W. hour is station costs.

If a hydro-electric company desires to sell its output to the steam electric company, it becomes evident that fixed charges are already incurred, that the only possible saving will be all or part of the station charges, as noted above. There is no immediate economy to the steam electric company in the purchase of water power generated current (although this may not be true if the future be taken into account), unless it can purchase it at less than its station cost; consequently, to effect a sale, the water power must be sold at from .8 cent to 1 cent per K. W. hour, delivered at the customer's switchboard. The higher price might probably be regarded as scarcely attractive to the steam electric company, and the lower price would seldom be attractive to the water power company, in many cases would involve a direct loss, and in almost every case a direct loss would be entailed unless the total amount of current sold to all customers constitutes a large proportion of the capacity of the plant. If a direct combination can be effected between the water power company and the steam electric company, the steam plant may be utilized as an auxiliary to the water power, and the whole value of the output utilized by the combined interests. This kind of a combination is usually the only way in which a larger net profit can be obtained from a water power development. The best results can be obtained only by combination with an industry or market already developed, in which the power can be utilized at its true market value.

CHAPTER XXIV

THE CONSIDERATION OF WATER POWER PROJECTS

340. Elements of Success.—The practicability of every water power project depends on a large number of factors each of which may be vital to its commercial success. Every factor, therefore, needs full and careful consideration before any attempt should be made toward the final consummation of the project. Many water power projects have proved disastrous financial failures. The mere fact that an undeveloped power is energy going to waste does not necessarily entail financial benefits when such energy is utilized by water power developments which may and frequently will involve large and unwarranted investments. Most water power projects are necessarily more or less speculative, that is the factors and their influence on the success of the project are more or less uncertain, and can not always be clearly determined. Frequently, the point of view of the promoter or investor is too optimistic. The difficulties are underestimated or ignored, and sometimes an uninformed public is induced to invest in projects which are undertaken without due consideration or in ignorance of the vital factors on which their success depends (see Chapter XXIII). Failures, partial or complete, have resulted from many causes, and a consideration of the principal ones which have occasionally given unsatisfactory results may serve to point out some of the dangers involved in these investments, and some of the points which should be given particular attention when such investments are seriously contemplated. Among the causes which have led to results more or less disastrous may be mentioned:

1. *The Market.*—The possibilities of the market for power are frequently overestimated. The actual market may be found inadequate. Occasionally water power plants are constructed to serve a prospective market such as a new industry which is proposed, either by the projectors of the water power project or by other interests, and the failure of the industry may result in a failure of the power project. A large investment based upon the success of another project controlled by other parties is usually a dangerous undertaking and should be attempted only after the greatest deliberation and most careful investigation. In many cases the market, while adequate, may already be sup-

plied by public utilities, and competition may involve such a reduction in the price of power that adequate returns can not be secured. Bond houses rightly look with distrust on investments which depend for their earnings on competition with plants already constructed and in successful operation and it is seldom possible to finance a project on such a basis. Frequently a market requires considerable time for its development. Interest begins with the first expenditure, and operating expenses usually reach well toward their maximum as soon as the plant begins operation. If the load is only slowly acquired, fixed charges and operating expenses will commonly exceed the income for a considerable period. The deficit must commonly be paid from the capital account, and if this condition long continues it may result in partial or complete failure. Only a few years of idleness involves either the extinction of the original investment or a large increase in the capital account; hence delays in acquiring a paying business are frequently disastrous.

2. *Water Supply.*—Success in water power projects depends to a large extent on the adequacy and constancy of the water supply. Such supplies are frequently overestimated, especially as to their constancy. In a few cases, this error has been so great that plants have been abandoned and dismantled. Occasionally the supply becomes so reduced in years of low rainfall that the resulting power output is found to be unsatisfactory. This condition usually involves either a low price for unsteady power which may result in unsatisfactory returns, or it may make necessary the installation of an auxiliary power plant which, if unforeseen, may involve overcapitalization or increased operating charges and consequent financial embarrassment. In this connection it is also important to fully appreciate the difference between the theoretical power of the stream and the actual power which can be delivered either at the turbine shaft or at the switchboard of the consumer, possibly many miles away. With modern and properly designed plants, it is possible to deliver at the turbine shaft from seventy-five to eighty per cent. of the theoretical energy of the water, even with a considerable variation in load, and even a higher percentage where the load is fairly constant. Where the installation involves transformation into electrical energy which must be stepped up to a high voltage for transmission purposes, and afterwards reduced to low voltage for secondary distribution and utilization, the percentage of delivered power will commonly be reduced, even under the most favorable conditions, to from sixty-five to sixty-eight per cent. of the theoretical power of the water,

and commonly even a greater loss must be sustained. Improper design may, of course, result in even greater losses which materially reduce the power output and the consequent income.

3. *Head.*—The reduction in head, in low head plants, due to high water conditions may result in occasional reduced output and make necessary the installation of auxiliary power, in order that constant service may be maintained. This, if unforeseen, involves additional expense which sometimes proves serious. This factor has commonly received even less attention than the variation in water supply, and in some cases may prove quite as serious.

4. *Cost.*—The cost of water power plants is frequently underestimated, and is perhaps the most common cause of failure, partial or complete. Excess cost is often due:

First: To unforeseen difficulties in the physical conditions, entailing extra costs of construction.

Second: To flood conditions during construction, and consequent extra expenses.

Third: To resulting delays and consequent excessive interest charges.

Fourth: To large and unexpected extra expenses for flowage, due to unforeseen backwater conditions.

Fifth: To over-development, or equipment not warranted by the market.

Sixth: To under-development on account of unforeseen low load factors.

Seventh: To unexpected cost of transmission due to difficulties in securing the necessary market.

Eighth: To expensive methods of financing.

Over development involves over capitalization, while under development requires an increased capitalization which will involve further and perhaps unwarranted investments.

5. *Maintenance.*—The hazards of maintenance have previously been considered (see Section 327, page 706). Maintenance may involve excessive cost due to unforeseen contingencies. The plant may be poorly designed, foundations may be inadequate or improperly protected, and other constructions may be unsuitable for their purposes. Many plants involve features and designs for which there is little precedent, and in such cases there is a necessary increased hazard.

The proper selection of machinery is an important feature of design. It must be efficient in order to develop maximum power. It must be of a suitable capacity in order that high efficiency will obtain. It must

be of a type needed to supply the market. A twenty-five-cycle hydroelectric development can not supply a sixty-cycle market without loss in efficiency and extra expense for installation. Hence an error in selecting machinery may involve replacement at large expense, and sometimes entail reconstruction at a still greater cost.

Poor workmanship frequently results in high maintenance costs, and rapid depreciation may be the result of poor construction. Floods are commonly under-estimated in volume, and when the maximum occurs it may result in partial or total destruction of the works. Even where due care has been taken to provide for passage of maximum floods, the failure of other works further up the stream, and perhaps entirely beyond the control of the engineer or proprietor of the water power plant, may result in unusual and serious catastrophes.

341. The Extent of Investigation.—From the previous discussion, it is evident that before a water power project is undertaken, a thorough study should be made of all of the factors involving its ultimate success. A market must be available where the power can be promptly sold at such prices that the returns will pay fixed charges, operating expenses, and a fair return on the investment. The water supply must be constant, and adequate, and the head available must be sufficient for the constant development of power, or the hydraulic plant must be supplemented by auxiliary power of such character and amount as will fill the market demands. The plant must be well and properly designed, and the machinery carefully selected with due regard to the maintenance of efficiency and market demands. The cost of construction must be reasonable so that the project will not be over capitalized. Operating and maintenance costs must be moderate.

All of these factors are more or less inter-dependent. One factor necessarily involves others to a greater or lesser extent and each factor must therefore be considered not only by itself but also in due relation to all others on which it may depend. Too often careful attention is given only to certain features of the project, and other features equally important are neglected or ignored. Preliminary to undertaking any project of this kind, a thorough examination should be made of all these elements, and a report based thereon should be prepared which should include a consideration of all factors and their relation to one another. Such a report should be made as thorough and as exhaustive as the project may demand. The expense involved in such an examination and report is fully warranted, and an investment in large undertakings of this character should never be enter-

tained without a complete and thorough investigation and a carefully considered report. The financial wrecks of a number of important projects point to the necessity of such an examination and illustrate the fact that it is either frequently omitted or improperly made. Dishonesty is undoubtedly an element in some of these failures but in most cases they are the result of carelessness, the neglect of a proper investigation or the lack of appreciation of all the elements involved.

The extent of the investigation must be governed by the importance of the project, and will also depend on whether the investigation and report are to be of a preliminary character or are to be the basis of a final report on which the feasibility of the project may be decided.

342. **Details of the Investigation.**—An investigation along the lines indicated must commonly be made by a number of experts, each familiar with the particular work entrusted to him. All such investigations, to be of full value, should be made under the direction of some one able to fully appreciate the necessary extent and character of such investigations, and to correlate and determine the influence of various factors and their mutual relations. The investigations necessary for a full understanding of the project and the procedures which should be followed, may be summarized in essentially their logical order as follows:

First: The determination of the market and the probable nature and distribution of the load. The present cost of power and the price at which power can be sold.

Second: The water supply and its variations. The possibilities of pondage and storage and the necessity and cost of auxiliary power.

Third: The physical character of the power site. The foundations, the plant layout, the resultant head, and local materials of construction, and the extent of the necessary transmission to convey the power to its market.

Fourth: The preliminary estimate of cost. This should be made very liberal and should contain estimates for contingencies amply sufficient to meet such uncertainties as are involved in the project.

Fifth: The legal rights, including riparian rights, flowage and reservoir rights; permits and franchises; rights-of-way for transmission; rights of condemnation; and the extent and influences of state and federal control on both project and prices.

Sixth: The methods and cost of financing the project, and the feasibility of floating the necessary stock and securities.

Seventh: A report on the project covering a full and complete consideration of Items 1 to 6, and their relations to one another.

Eighth: The design of the plant, and the structures necessary for storage and control of waters.

Ninth: The construction.

Tenth: Operation and maintenance.

343. Market or Purpose of Development.—Every water power project must be examined in the light of the purpose for which it is to be used, or the market which it is to supply. In some cases cheap power is more essential than steady power and if ample power can be furnished for a large portion of the year, inadequate supplies during low water and limited head during flood conditions may be overlooked if, for the balance of the year, the power can be furnished at a low rate. In most cases, however, where the power is to be sold for use in various industries, the water supply must be constant and continuous not only for every day in the year, but for every day of every year of its operation; and a large reduction in power during drought, high water or other contingencies, that will temporarily suspend or materially reduce the output of power by the plant, will not be satisfactory. In such cases, the development can only be carried to the extent that the minimum flow will warrant, or auxiliary power must be provided to furnish the deficiency whenever it occurs. When the market demand is continuous, in order to avoid loss or great inconvenience, precautions must be taken to so design the plant with duplication of parts, extra units, suitable pondage or storage, and such sufficient auxiliary sources of power that all interruptions shall be practically eliminated.

Usually a census should be made of the power actually in use in the territory to be served. An accurate estimate of the amount of power used by a factory or manufacturing plant is a matter of considerable difficulty. In some plants where power is electrically distributed, the use of indicating, and sometimes of recording instruments, make it very easy to determine the energy output of the power plant. In most manufacturing establishments where power is distributed by belts, shafting, and other than electrical means, the amount of power actually developed and utilized is seldom definitely known. The use of the steam engine indicator, if opportunity for such use is offered, will give a knowledge of the indicated power of the engine at the time observations are made; and if the probable variations are investigated, a fairly close estimate of power used can often be made by this means.

The annual quantity of coal used, and a careful study of the condition and character of the boiler service, requirements for heating, con-

dition of the engine used, together with a careful examination of the machinery operated, will form the basis of a fairly approximate estimate of power used. Even where the estimate of power used is fairly accurate, it must be remembered that when such power is used and transmitted through a multitude of shafts, belts, etc., that if the electric power is substituted and individual motors used on the machine to be operated, the power then used will be very greatly reduced in amount.

Occasionally the use of exhaust steam for heating or manufacturing purposes makes power generated by steam a by-product during certain portions of the year, and may materially reduce the price which a manufacturer can pay for power generated by water. Sometimes the necessity of maintaining power at all times or the necessity of providing for low stages of river flow, makes it desirable to still maintain the steam plant even though power from water is utilized, in which case also the price which can be economically paid for water power will be considerably reduced.

Usually the operating costs are the upper limit of the charges which can be made where power users are to be supplied from the new source, and frequently a considerable reduction below this cost is necessary in order to induce power users to abandon old methods and adopt a new source of power which will involve a considerable extra expense for a new installation. The distribution of the load or the probable load curve which can be developed has a marked influence on the nature and extent of the development and equipment. Whether power is to be used for ten-hour, twelve-hour or twenty-four-hour service, and its probable variation during the hours of use, has a most important bearing on the design and cost of the plant (see Chapter VII). If variations in the demand for power throughout the year are likely to occur, such variations will affect the requirements for storage and the maximum possible output of the plant, and must receive due consideration.

344. Preliminary Investigation of Water Supply.—An examination of the data available in any first class engineering library will generally give the information necessary to form an approximate judgment of the probable feasibility of a project in so far as it depends on the flow of the stream. The approximate area drained by the stream can be determined by reference to such maps as may be available and the probable flow and the variations in the same that will occur from day to day and from month to month, can usually be approximately

determined by the construction of comparative hydrographs made from either the measured flow of the stream, if such information is available (see Literature page 166) or otherwise on the comparative flow of similar and adjacent streams as described in Section 72, page 143.

From such an investigation together with an appropriate knowledge of the available head, an estimate of the probable power of the stream that can be obtained and maintained can be made, and from such information an opinion can be formed as to whether it is desirable to carry the investigation further. Frequently such an investigation will show beyond question the futility of the project, and even an examination of the locality becomes unnecessary. If the preliminary investigation shows that sufficient power is probably available on the stream in question the investigation can be carried into sufficient detail to warrant an opinion as to whether or not the project is feasible in all of its phases.

345. **Study of Stream Flow.**—The information of primary importance in a water power project is the amount and variation in the run-off of the stream itself. If this is not available the run-off of neighboring streams that have similar physical and meteorological conditions prevailing on their drainage area is next in importance.

As already pointed out (see Section 71, page 141), the hydrograph of the actual flow of the stream itself is the best information for studying its variations in flow. Such hydrographs to be conclusive must be available for a considerable term of years, and it is desirable that they should cover all extremes of rain-fall, drought, temperature and other meteorological and physical conditions that influence run-off.

In the investigation of the hydrographical condition of any stream, a single gauging of the stream is of little or no value. It is however, desirable to establish a gauging station as early as possible and to take daily gauge readings. It is also important, both for the purpose of an understanding of the gauge reading and for the purpose of the study of head, to make stream flow measurements (see Chapter V) under all large variations in flow, as early as possible in order that a rating curve may be established.

When no local hydrographs are available, or when such available hydrographs are limited to a few years, it becomes desirable to gather together the flow data of all adjacent and similar streams and to construct comparative hydrographs therefrom, as described in Section 72, page 143. A long continued series of hydrographs of a neighboring stream where similar conditions prevail is important and should usually be utilized

even if local observations have been made for a few years. The value of comparative hydrographs is dependent on the similarity of conditions, a question that demands careful consideration and a considerable amount of data to determine, and even then can be regarded only as indicative. It is also essential to make careful comparisons of the relations that exist between the hydrograph of the river under discussion and those of adjoining rivers, for such period as the data may be mutually available on both streams, in order that variations between the areas compared may be determined.

346. **Study of Rain-Fall.**—The rain-fall records of the United States Weather Bureau and, previous to these, the records of the observations of the United States Signal Service are available from various stations throughout the United States for a long term of years. It is often desirable to collect and to study the rain-fall data for the drainage area of the stream under consideration, and also on such other drainage areas as may be used for comparative purposes. In investigating rain-fall it is usually particularly desirable to make a study of both the annual rain-fall and for the periodical rain-fall since the distribution of the rain-fall frequently has a greater effect on the low water flow than the total rain-fall for the year.

The relations between rain-fall and run-off for the period for which complete data is available should be investigated and such relations established as clearly as possible for the drainage areas under consideration. While such relations are variable and more or less indefinite, they are frequently indicative and an aid to judgment, especially in relation to extreme conditions. With the information concerning run-off commonly available and the rain-fall records for a considerable term of years, it will be possible to draw fairly accurate conclusions as to the probable variation and average flow of the stream. The probability of a larger maximum or a smaller minimum than the stream flow observations themselves indicate can also be determined from such an investigation.

347. **Study of Topographical and Geological Conditions.**—The topographical and geological conditions may ordinarily be investigated from data available in the publications of the United States Geological Survey, or of the geological surveys of the state in which the drainage area may lie. The information sought from this investigation is a knowledge of the conditions that will effect run-off, consequently, such a study is not of particular importance provided sufficient rain-fall and run-off data is available for the purpose of the investigation.

If, however, the hydrographical condition of the areas under consideration, or of other adjacent and similarly located areas have not been previously investigated, and if few or no local observations of stream flow have been made, the topographical and geological data may form the basis of a more intelligent opinion in regard to the probable run-off than can be obtained without such consideration. In any event, this source of information should be utilized to the full extent warranted, as should all other sources of information that will in any way assist the engineer in an intelligent understanding of the problem before him, and the formation of a correct opinion as to the possibilities and probabilities of the case in question.

348. **Study of Flood-Flow.**—It is important to establish both from information that is usually available in the stream valley under consideration, and from information which may be available from adjoining streams, the probable maximum flood-flow of the stream. This must be determined, or at least a safe approximate estimate must be made in order that the dam and other works for the control of the flow can be intelligently designed (see Section 299, page 612).

The flood-flow is a condition which needs the most careful consideration for it is often the condition of greatest danger and, to assure safe construction during the short period when such conditions obtain, requires special attention and intimate knowledge of the local conditions, and often involves considerable expense.

After the rating curve has been established the elevation of the high water marks in the immediate vicinity and the relation of the same to guage heights will usually give a safe basis for the estimate of extreme flood-flows, but a considerable factor of safety should usually be provided for the exceptional condition which may not have been included in the limited records available. The recent (1915) occurrence of serious and unprecedented floods in various parts of the United States emphasizes the fact that maximum floods are apt to occur at long intervals and under exceptional conditions, and that limited records will give little indication of their possible magnitude. This matter therefore is worthy of the most complete investigation and thoughtful consideration.

349. **Study of the Back-Water Curve.**—A topographical survey of the proposed site of the dam and of the stream valley above the dam site, to the probable practical limit of the back-water effect, should be made. In order to investigate the probable height of the backwater under all conditions of flow it will be necessary to make cross-

sections of the river at such intervals and under such conditions as will permit of the division of the river into lengths or reaches having comparatively uniform sections. Gauges should then be established at the various stations and observations should be made of the gauge heights at each station during various stages of flow (see Section 47, page 87). From the quantity of water flowing at any stage, together with the cross-sections of the river on the various divisions, the value of the hydraulic elements and especially of the friction coefficients for each division and their variations under such condition of flow, can be calculated (see Sections 36 to 51, pages 71 to 101). After this has been done it is possible to calculate the back-water curve (see Section 51, page 101) and to establish the probable limit of the back-water flow line under any other conditions of flow in a fairly reliable manner.

350. Study of Head.—The consideration of these conditions, the height of the water surface at the dam due to various sections and length of the spill-way and the practicable limit to which flood height in the valley above must be restricted, will usually establish the limit of the height to which the dam can or should be built and will indicate whether it is necessary or desirable to construct flood gates or to use an adjustable crest, flash boards, or other means for regulating and limiting the flood height. When these conditions are established the variations in head under various conditions of flow can be determined (see Chapter VIII, page 186) and the effect of such variations on the power which may be developed can be calculated (see Section 96, page 193).

351. Study of Storage and Pondage.—The topographical survey will also give information concerning the storage and pondage condition immediately above the dam. In special cases, reservoirs beyond the limit of the back-water effect may be desirable and special surveys under such conditions will be necessary. As the conditions of pondage and storage materially affect the amount of power available, these questions frequently become of great importance and should receive the attention of the engineer that their importance in each particular case seems to warrant. After definite information is obtained concerning the extreme permissible limit of flood-flow, and the possibilities of storage and pondage, an estimate of the power of the stream under various conditions of use can be readily made (see Chapter VII, page 168). It is important in considering the power of the stream and especially the desirable condition of pondage, to ascertain as far as

practicable the probable necessary distribution of the demand for power throughout the day (see Section 79, page 168).

352. **Physical Investigations.**—The physical and geological condition which will influence the design and necessary character of the construction to be installed should be carefully investigated. It is necessary to examine in considerable detail the bed and banks of the stream and to make more or less extensive soundings in order to establish all conditions on which the character of the construction to be recommended must depend. It is important that all conditions be carefully investigated and the type of construction to be recommended be carefully considered. The storage of energy almost always involves a hazard which must be met with economical but safe design and construction. The prevention of flow under and around the structure requires a detailed knowledge of the local conditions and is one of the most uncertain conditions which, unless carefully and correctly estimated, is apt to result in considerable extra expense. The cost of a thorough investigation of these conditions is usually a considerable item of expense which investors hesitate to authorize until it is decided to undertake the installation. Estimates made without such an investigation are subject to grave errors, and failure to undertake this necessary study is the most frequent cause of underestimates.

353. **Study of Power Development.**—Having established the probable load curve, the head under all conditions of flow, and the flow as modified by the pondage or storage conditions, the extent of the power development can be determined. All of the questions that have been previously discussed lead up to the consideration of the question of the desirable capacity or extent of the proposed power development. This capacity should always be estimated on a conservative basis. If, as is usually the case, uncertainties exist as to the probable demand and distribution of power, or the probable minimum flow of the stream, it is desirable to develop the project to a point below the probable commercial maximum but to keep in mind the possibility of future extensions and to consider the plans with the future in view. In this connection the question of auxiliary power, and the capacity of the plant as modified by such power, should receive attention.

354. **Study of Auxiliary Power.**—The necessity of auxiliary power in connection with the proposed water power development can be determined by an intelligent study of the hydrograph and an investigation of the effects thereon of the storage and pondage available (see Section 84, page 173). As a general principle, it may be stated that a

stream can often be developed to commercial advantage to the extent of the power which will be uniformly available for eight or nine months of the driest year, and that auxiliary power is usually warranted to furnish the power needed for the remainder of the season. This is a general rule which must be applied with caution. Every proposed development must be carefully investigated for itself, and no general conclusion should form the basis of final opinion on the feasibility of such a project. The greater the demand for power, and the greater the cost of development from other than water power sources, the more the expense which is warranted for auxiliary service, pondage, etc., and the greater the capacity to which the water power can be utimately developed.

355. **The Estimate of Cost.**—In order that the preliminary estimate shall be made on a safe basis, reasonable allowances should be made for unforeseen and possible contingencies. This is especially desirable in preliminary estimates on which the feasibility of the entire project may be based. If a safe estimate of the probable cost of construction,—that is an estimate which will surely not be exceeded and will probably be reduced in construction,—makes the feasibility of the project doubtful, then, as a general proposition, the project is not worthy of further consideration. If the project is predicated on the basis of an estimate that is known to be safe, it can lead to no unfortunate investments. The owners of a development are always satisfied if the ultimate cost is less than the engineer's estimate; but an increase in cost is often a serious matter.

The desire to proceed with the development of a project is sometimes apt to give an optimistic coloring to the engineer's report. This is a tendency which, both on account of the interest of his client and his own future reputation, he should carefully avoid.

If the feasibility of the project is reasonably well established by the preliminary examination, the investigation should be still further extended and made complete. Preliminary plans should be outlined in order that a safe detailed estimate may be made. The expense involved in such preliminary work is well warranted by the results obtained. In many cases plants have been recommended on insufficient examination, and the estimates made with too optimistic a view of the conditions to be met. The latter development of the necessity of increased expense, has made the project less attractive and has resulted in great disappointment both to the owners and to the engineer on whose opinion the work has been undertaken.

356. Legal Rights.—Every phase concerning the legal rights which must be acquired should receive due consideration. In many states water power development has been hampered by unjust and restrictive laws which may seriously affect the commercial value of any water power development (see Section 13, page 21). The federal and state laws affecting the acquisition of riparian rights, flowage, reservoir sites and construction, and the rights-of-way for transmission purposes, require investigation. The state and federal laws, passed to control these projects, limit the prices at which power can be sold, and the ultimate disposition of the plant should also be investigated. The practicability of the project frequently depends upon the rights of condemnation for flowage and reservoir purposes, and unless these rights can be acquired by eminent domain, the acquisition of the necessary lands will frequently necessitate a large and unwarranted, and even a prohibitive expense.

357. Financing.—The methods and cost of financing, together with certain legal and business aspects of the questions involved, have already been discussed at some length in Chapter XXII. These methods must be considered with the restrictions imposed by the state and federal laws, which restrictions frequently prove so onerous as to make developments impossible.

358. The Report.—As far as practicable, a report on a water power project, should include all of the data on which deductions are based. This data and its bearings on the project should be fully discussed and the reasons for the opinions expressed should be clearly set forth. In a well drawn report the engineer can usually so illustrate and describe the conditions by which a project is modified and controlled that any good business man will understand the basis on which the opinion rests and the degree of probability of any departure from the expected result. While this is not true in regard to the technical details, it is entirely true of the general consideration on which the feasibility of a project depends. If a report can not be so drawn it is due either to insufficient data or to the fact that the engineer himself does not fully understand and appreciate the logic of the situation.

In general, a complete report on a water power project should include a careful consideration and discussion of the following:

First: A general description of the drainage area, including the size and the topographical, geological, and other physical conditions that may have a direct bearing on the feasibility of the project.

Second: The run-off data available on the streams in question, if any such data exists.

Third: If local run-off data is available for a brief term of years only, the rain-fall of the district for as long a period as possible should be collected, and its relations to the available run-off data established. From this the probable modification of the run-off during other years during which the rain-fall is found to vary, should be carefully and fully discussed.

Fourth: The run-off data on adjoining streams, having drainage areas with similar physical, topographical and geological conditions, and where the hydrographical conditions of the rain-fall and run-off are apparently similar, when the difference therein can be determined and estimated should be graphically presented and discussed.

Fifth: The relations of the rain-fall and of other conditions on the comparative areas considered, and their variations from the particular location under consideration, should be fully illustrated.

Sixth: The conclusion in regard to the probable flow from the drainage area, considered on the basis of its run-off, and the run-off of comparative areas should be fully considered.

Seventh: A general description should be given of the localities at which the dam and power stations are to be constructed; the physical conditions there existing, and the effect of such conditions upon the construction of the plant and the methods of meeting them, should be fully outlined. There should also be furnished a description of the work proposed, sufficiently complete to form a basis for estimates of probable cost, depreciation and fixed charges, cost of maintenance and of operation.

Eighth: The head available and the variations under various conditions of flow should also receive careful consideration.

Ninth: The probable power available with and without pondage, or with the pondage found to be available by preliminary survey should be fully treated. The probable output in horse power hours or kilowatt hours which can be developed and sold, with unit cost of power base on fixed and operating charges, should be estimated.

Tenth: The auxiliary power, if any, necessary to maintain the plant at all times to the capacity recommended needs specific discussion, and an estimate should be made of the cost of the necessary installation and the effect of such cost on the cost of the total power output.

Eleventh: An estimate should be made of the probable cost of the development, the probable fixed charges. operating expenses, and cost of maintenance.

Twelfth: The probable market for the power to be generated, the probable distribution of the demand for the power through the day and year, and the basis on which such estimates are made, should be given.

Thirteenth: The sources of power used or which may be used in the territory which it is proposed to supply, the cost of developing the same, also the probable price per unit of energy at which the power can be sold immediately and ultimately, the probable income therefrom and the availability to meet fixed charges, maintenance, operating expenses and profit, should be fully set forth.

Fourteenth: The report should be accompanied by hydrographs and such other drawings as will, with the data furnished, show conclusively that the facts are as the report sets forth.

Fifteenth: The national and state laws affecting the organization, acquisition of rights and privileges, operation and the sale of power should be discussed.

Sixteenth: In general, it is advisable that the report itself should be clear, concise, and definite in its statements and recommendations. Any elaborate discussion of voluminous data should be furnished in the form of an appendix to which the main report should refer for confirmation of its findings and recommendations.

359. Study of Plant Design.—The study of plant design requires an extensive study of the various types of development that are in practical use and the adaptability of such designs to the conditions of the particular locality under consideration. It is seldom that plans, no matter how successfully carried out in one place, can be duplicated to advantage in another. Each plant should be built to meet the particular conditions under which it is to be installed and operated, and the best ideas from all sources that will apply to the local conditions should be correlated and embodied in the proposed plant. Extensive experience, observation, and study are each desirable and essential for the best results. For his own, as well as for his client's good, the engineer should endeavor to secure the very best results possible when all things are carefully weighed and considered. No reasonable amount of conscientious work, painstaking thought, study, labor or expense should stand in the way of such results; and anything less than this is a detriment to future professional attainments which no engineer, young or old, can afford.

In the previous chapters the general principles underlying the design of the various elements of the plant have been considered. The

consideration of these matters has been very brief and the engineer must extend his study in all cases to the extensive literature on each subject, reference to some of which has been given at the end of most chapters. Additional references can be found in the Engineering Index and in the indexes to the various technical publications and the publications of the various engineering societies. A personal visit to and a detailed examination of successful plants is a method for the acquisition of exact knowledge which should not be neglected. New, novel and untried designs are frequently described in engineering publications. If they are successful their success is often heralded in a similar manner. Their failure is seldom mentioned by the technical press and the only method of ascertaining their true value is by personal and confidential inquiry on the ground.

360. **Construction.**—In the construction of water power plants, great care should be exercised to obtain the best class of workmenship. Hydraulic work is perhaps the most difficult work of construction. A large element of cost is entailed in the building of cofferdams and in the unwatering of foundations, and success in this line is assured only by extensive experience and can usually be acquired in no other way. After these prelimiary expenses are incurred, the greatest care should be exercised to secure the very best results, especially in subaqueous work, for the repair or renewal of such work may frequently involve great extra cost, the necessity of which may be entirely eliminated by care in the original construction. Construction work should therefore be placed in the hands of competent, skillful, experienced and reliable contractors, and should receive supervision of a similar character.

361. **Operation and Maintenance.**—The success of the plant may be greatly modified by excessive costs in operation and maintenance. These costs are largely controlled by care in design and in construction. The plans on which the plant is constructed should be so modified as to show the actual construction as finally installed and should be kept for all time so that a full knowledge may be available in case repairs, reconstruction or extensions are to be made. In the course of operation, it is also important that complete records be kept of all data of operation, including the constant variations of flow in the stream, and especially of the minimum and maximum flows which occur only occasionally. Such information may be of little value in the daily operation of the plant, but when extensions or reconstruction becomes necessary, they are of the utmost importance as a satisfac-

tory basis for design. Unless accurately kept for a long term of years, these records are of little value.

Those in continuous charge of a plant are very apt to overlook many conditions in operation and maintenance, because these conditions develop only slowly. It is therefore exceedingly important that the plant, its equipment, operation and maintenance, be carefully reviewed at least once a year by some expert thoroughly and practically familiar with such matters. Constant familiarity with a plant entails a loss of perspective by those in constant charge, and an expert less familiar with the gradual changing conditions may frequently make suggestions as to economy of operation and necessary provisions of safety that may be overlooked by the regular operating force.

APPENDIX A

MISCELLANEOUS TABLES.

TABLE 57.

Table of three-halves ($\frac{3}{2}$) powers of numbers from 11 to 20 feet, varying by one-hundredth foot.

	11	12	13	14	15	16	17	18	19	20
.00	36.48	41.57	46.87	52.39	58.09	64.00	70.09	76.37	82.82	89.44
.01	.53	.62	.93	.44	.16	.06	.15	.43	.88	.51
.02	.58	.67	.98	.50	.21	.12	.21	.50	.95	.58
.03	.63	.72	47.03	.55	.27	.19	.27	.56	83.01	.65
.04	.68	.78	.09	.61	.33	.24	.33	.62	.08	.72
.05	.73	.82	.14	.66	.39	.30	.40	.69	.14	.78
.06	.78	.88	.20	.72	.45	.36	.46	.75	.21	.85
.07	.83	.93	.25	.77	.51	.42	.52	.82	.27	.92
.08	.88	.98	.31	.83	.56	.48	.58	.88	.33	.99
.09	.93	42.04	.37	.89	.62	.54	.64	.94	.40	90.06
.10	.98	.09	.42	.95	.68	.60	.71	77.01	.47	.12
.11	37.03	.14	.47	53.01	.74	.66	.77	.07	.53	.18
.12	.08	.20	.53	.06	.80	.72	.83	.13	.60	.25
.13	.13	.25	.58	.12	.85	.78	.89	.20	.66	.32
.14	.18	.30	.64	.17	.91	.84	.95	.26	.73	.39
.15	.23	.35	.69	.23	.97	.90	71.02	.32	.79	.46
.16	.28	.40	.74	.28	59.03	.96	.08	.39	.85	.53
.17	.33	.45	.79	.34	.09	65.02	.14	.45	.92	.60
.18	.38	.51	.85	.40	.15	.08	.20	.51	.98	.66
.19	.43	.56	.90	.45	.21	.14	.26	.58	84.07	.73
.20	.48	.61	.96	.51	.26	.20	.33	.64	.13	.79
.21	.53	.67	48.01	.57	.32	.26	.39	.70	.19	.85
.22	.58	.72	.07	.62	.38	.32	.45	.77	.26	.92
.23	.63	.77	.12	.68	.44	.38	.52	.83	.32	.99
.24	.68	.82	.17	.73	.50	.44	.58	.90	.39	91.06
.25	.73	.87	.23	.79	.56	.50	.64	.96	.45	.13
.26	.78	.93	.29	.85	.61	.56	.71	78.03	.52	.20
.27	.83	.98	.34	.91	.67	.63	.77	.09	.58	.26
.28	.88	43.03	.40	.96	.73	.69	.83	.16	.65	.33
.29	.93	.09	.45	54.01	.79	.75	.89	.22	.72	.40
.30	.98	.14	.51	.07	.85	.81	.96	.29	.79	.46
.31	38.03	.19	.56	.13	.91	.87	72.02	.35	.86	.53
.32	.09	.24	.62	.19	.97	.94	.08	.41	.92	.60
.33	.14	.30	.67	.25	60.02	.99	.14	.48	.99	.67
.34	.19	.35	.72	.30	.08	66.05	.20	.54	85.05	.73
.35	.24	.40	.79	.36	.14	.11	.27	.60	.12	.80
.36	.29	.45	.84	.42	.20	.17	.33	.67	.18	.87
.37	.34	.51	.89	.48	.26	.24	.39	.73	.25	.94
.38	.39	.56	.94	.54	.32	.30	.45	.80	.32	92.00
.39	.44	.61	49.00	.59	.38	.36	.51	.87	.38	.07
.40	38.49	43.67	49.06	54.65	60.44	66.42	72.58	78.93	85.45	92.14
.41	.54	.71	.11	.70	.50	.48	.64	.99	.51	.20
.42	.59	.77	.16	.76	.56	.54	.70	79.06	.58	.27
.43	.64	.83	.22	.82	.61	.60	.77	.12	.64	.34
.44	.69	.87	.27	.87	.67	.66	.83	.19	.71	.40

TABLE 57—Continued.

	11	12	13	14	15	16	17	18	19	20
.45	.74	.93	.33	.93	.73	.72	.89	.25	.77	.47
.46	.79	.98	.38	.99	.78	.78	.96	.32	.84	.54
.47	.84	44.04	.44	55.04	.84	.84	73.02	.38	.90	.61
.48	.89	.09	.50	.10	.90	.90	.08	.44	.97	.68
.49	.94	.14	.55	.16	.96	.96	.15	.51	86.04	.75
.50	39.00	.20	.60	.22	61.02	67.03	.21	.57	.11	.82
.51	.05	.25	.66	.27	.08	.09	.27	.63	.17	.89
.52	.10	.30	.71	.33	.14	.15	.33	.70	.23	.96
.53	.15	.36	.77	.39	.20	.21	.40	.76	.30	93.03
.54	.20	.41	.82	.44	.26	.27	.46	.83	.36	.10
.55	.25	.46	.88	.50	.32	.33	.52	.89	.42	.16
.56	.30	.51	.94	.56	.38	.39	.59	.96	.49	.23
.57	.35	.57	.99	.62	.44	.45	.65	80.02	.56	.30
.58	.40	.62	50.04	.67	.50	.51	.71	.09	.63	.36
.59	.45	.67	.10	.73	.56	.58	.78	.15	.70	.43
.60	.51	.73	.15	.79	.62	.64	.84	.22	.77	.50
.61	.56	.78	.21	.85	.68	.70	.90	.28	.84	.57
.62	.61	.83	.27	.90	.74	.76	.96	.35	.91	.64
.63	.66	.89	.32	.96	.80	.82	74.03	.41	.97	.70
.64	.71	.94	.38	56.02	.85	.88	.09	.48	87.04	.77
.65	.76	.99	.43	.08	.92	.94	.15	.54	.11	.84
.66	.81	45.05	.48	.13	.97	68.00	.22	.61	.17	.90
.67	.86	.10	.54	.19	62.03	.07	.28	.67	.23	.97
.68	.91	.15	.60	.25	.09	.13	.34	.74	.30	94.04
.69	.96	.20	.65	.30	.15	.19	.41	.80	.37	.11
.70	40.02	.26	.71	.36	.21	.25	.47	.87	.44	.18
.71	.07	.31	.77	.42	.27	.31	.53	.93	.51	.25
.72	.12	.37	.82	.48	.33	.38	.59	81.00	.58	.32
.73	.17	.42	.88	.54	.39	.44	.66	.06	.64	.38
.74	.22	.47	.94	.59	.45	.50	.72	.13	.71	.44
.75	.27	.53	.99	.65	.51	.56	.78	.19	.78	.51
.76	.32	.58	51.04	.71	.57	.62	.85	.26	.84	.58
.77	.37	.63	.10	.76	.63	.68	.91	.32	.91	.64
.78	.42	.69	.16	.82	.69	.74	.97	.39	.98	.71
.79	.47	.75	.21	.88	.75	.80	75.04	.45	88.04	.78
.80	40.53	45.80	51.26	56.94	62.80	68.86	75.10	81.52	88.10	94.86
.81	.58	.85	.32	57.00	.86	.92	.16	.58	.17	.93
.82	.63	.91	.38	.05	.92	.98	.22	.65	.24	95.00
.83	.68	.96	.43	.11	.98	69.05	.29	.71	.30	.07
.84	.74	46.01	.48	.17	63.04	.11	.35	.78	.37	.14
.85	.79	.07	54	.23	.10	.17	.41	.84	.44	.21
.86	.84	.12	.60	.29	.16	.23	.48	.91	.50	.27
.87	.89	.18	.66	.34	.22	.29	.54	.97	.57	.34
.88	.95	.23	.71	.40	.28	.35	.60	82.04	.64	.41
.89	41.00	.28	.77	.46	.34	.41	.67	.10	.71	.48
.90	.05	.33	.82	.52	.40	.48	.73	.17	.77	.55
.91	.10	.39	.88	.58	.46	.54	.79	.23	.84	.62
.92	.15	.44	.93	.63	.52	.60	.85	.30	.91	.69
.93	.20	.50	.99	.69	.58	.66	.92	.36	.97	.76
.94	.26	.55	52.05	.74	.64	.72	.98	.43	89.04	.82
.95	.31	.61	.11	.81	.70	.78	76.05	.49	.11	.89
.96	.36	.66	.16	.86	.76	.85	.11	.56	.17	.96
.97	.41	.71	.21	.92	.82	.91	.17	.62	.24	96.03
.98	.47	.77	.27	.98	.88	.97	.24	.69	.31	.10
.99	.52	.82	.33	58.04	.94	70.03	.30	.75	.37	.17

TABLE 58.

Table of three-halves ($\frac{3}{2}$) power of numbers from 0 to 100 feet, varying by one-tenth foot.*

Head in feet.	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9
0....	0.0000	0.0316	0.0894	0.1643	0.2530	0.3536	0.4648	0.5857	0.7155	0.8538
1....	1.0000	1.1537	1.3145	1.4822	1.6565	1.8371	2.0238	2.2165	2.4150	2.6190
2....	2.8284	3.0432	3.2631	3.4881	3.7181	3.9529	4.1924	4.4366	4.6853	4.9385
3....	5.1962	5.4581	5.7243	5.9947	6.2693	6.5479	6.8305	7.1171	7.4076	7.7019
4....	8.0000	8.3019	8.6074	8.9167	9.2295	9.5459	9.8659	10.1894	10.5163	10.8466
5....	11.1803	11.5174	11.8578	12.2015	12.5485	12.8986	13.2520	13.6086	13.9682	14.3311
6....	14.6969	15.0659	15.4379	15.8129	16.1909	16.5718	16.9557	17.3425	17.7322	18.1248
7....	18.5203	18.9185	19.3196	19.7235	20.1302	20.5396	20.9518	21.3666	21.7842	22.2045
8....	22.6274	23.0530	23.4812	23.9121	24.3455	24.7815	25.2202	25.6613	26.1050	26.5523
9....	27.0000	27.4512	27.9050	28.3612	28.8199	29.2810	29.7445	30.2105	30.6789	31.1496
10....	31.6228	32.0983	32.5762	33.0564	33.5390	34.0239	34.5111	35.0006	35.4924	35.9865
11....	36.4829	36.9815	37.4824	37.9855	38.4908	38.9984	39.5082	40.0202	40.5343	41.0507
12....	41.5692	42.0910	42.6128	43.1388	43.6648	44.1952	44.7256	45.2600	45.7944	46.3332
13....	46.8720	47.4148	47.9576	48.5048	49.0520	49.6032	50.1544	50.7096	51.2648	51.8240
14....	52.3832	52.9464	53.5096	54.0768	54.6440	55.2152	55.7864	56.3616	56.9368	57.5156
15....	58.0944	58.6776	59.2608	59.8472	60.4336	61.0244	61.6152	62.2096	62.8040	63.4020
16....	64.0000	64.6020	65.2040	65.8096	66.4152	67.0244	67.6336	68.2464	68.8592	69.4760
17....	70.0928	70.7132	71.3336	71.9572	75.5808	73.2084	73.8360	74.4672	75.0984	75.7328
18....	76.3672	77.0056	77.6440	78.2856	78.9272	79.5724	80.2176	80.8664	81.5152	82.1672
19....	82.8192	83.4748	84.1304	84.7892	85.4480	86.1104	86.7728	87.4384	88.1040	88.7732
20....	89.4424	90.1152	90.7880	91.4636	92.1392	92.8184	93.4976	94.1800	94.8624	95.5484
21....	96.2344	96.9232	97.6120	98.3044	98.9968	99.6924	100.3880	101.0868	101.7856	102.4872
22....	103.1883	103.8940	104.6008	105.3076	106.0160	106.7276	107.4392	108.1540	108.8688	109.5864
23....	110.3040	111.0248	111.7456	112.4700	113.1944	113.9216	114.6488	115.3788	116.1088	116.8420
24....	117.5752	118.3128	119.0496	119.7876	120.5272	121.2696	122.0120	122.7576	123.5032	124.2516
25....	125.0000	125.7516	126.5032	127.2576	128.0120	128.7706	129.5292	130.2876	131.0480	131.8112
26....	132.5744	133.3408	134.1072	134.8764	135.6456	136.4180	137.1904	137.9652	138.7400	139.5180
27....	140.2960	141.0768	141.8576	142.6416	143.4256	144.2120	144.9984	145.7880	146.5776	147.3700
28....	148.1624	148.9572	149.7520	150.5500	151.3480	152.1488	152.9496	153.7532	154.5568	155.3632
29....	156.1696	156.9788	157.7880	158.6000	159.4120	160.2268	161.0416	161.8588	162.6760	163.4964
30....	164.3168	165.1396	165.9624	166.7884	167.6144	168.4428	169.2712	170.1020	170.9328	171.7668
31....	172.6008	173.4372	174.2736	175.1128	175.9520	176.7940	177.6360	178.4804	179.3248	180.1720
32....	181.0192	181.8692	182.7192	183.5716	184.4240	185.2792	186.1344	186.9920	187.8496	188.7100
33....	189.5704	190.4336	191.2968	192.1624	193.0280	193.8960	194.7640	195.6348	196.5056	197.3788
34....	198.2520	199.1460	200.0400	200.9008	201.7616	202.6424	203.5232	204.4068	205.2904	206.1764
35....	207.0624	207.9512	208.8400	209.7312	210.6224	211.5204	212.4184	213.3104	214.2024	215.1012
36....	216.0000	216.9012	217.8024	218.7060	219.6096	220.5760	221.4224	222.3312	223.2400	224.1512
37....	225.0624	225.9760	226.8896	227.8056	228.7216	299.6404	230.5592	231.4800	232.4008	233.3244
38....	234.2480	235.1736	236.0992	237.0276	237.9560	238.8868	239.8176	240.7508	241.6840	242.6196
39....	243.5552	244.4932	245.4312	246.3712	247.3112	248.2540	249.1968	250.1420	251.0872	252.0348
40....	252.9824	253.9320	254.8816	255.8340	256.7864	257.7412	258.6960	259.6528	260.6096	261.5688
41....	262.5280	263.4896	264.4512	265.4152	266.3792	267.3456	268.3120	269.2804	270.2488	271.2200
42....	272.1912	273.1644	274.1376	275.1132	276.0888	277.0672	278.0456	279.6252	280.0048	280.9872
43....	281.9696	282.9544	283.9392	284.9264	285.9136	286.9028	287.8920	288.8836	289.8752	290.8692
44....	291.8632	292.8597	293.8552	294.8536	295.8520	296.8528	297.8536	298.8564	299.8592	300.8640
45....	301.8688	302.8764	303.8840	304.8936	305.9032	306.9148	307.9264	308.9404	309.9544	310.9708
46....	311.9872	313.0056	314.0240	315.0448	316.0656	317.0877	318.1112	319.0556	320.0000	321.1080
47....	322.2160	323.2452	324.2744	325.3060	326.3376	327.3716	328.4056	329.4416	330.4776	331.5156
48....	332.5536	333.5927	334.6333	335.4753	336.7188	337.7588	338.8051	349.8529	340.8972	341.9479
49....	343.0000	344.0486	345.0986	346.1500	347.2079	348.2622	349.3179	350.3750	351.4336	352.4886
50....	353.5500	354.6128	355.6720	356.7376	357.7996	358.8681	359.9329	360.9992	362.0719	363.1409
51....	364.2114	365.2832	366.3564	367.4311	368.5020	369.5794	370.6582	371.7333	372.8149	373.8927
52....	374.9772	376.0578	377.1397	378.2331	379.3078	380.3940	381.4815	382.5703	383.6606	384.7522
53....	385.8453	386.9343	388.0301	389.1219	390.2205	391.3150	392.4163	393.5136	394.6122	395.7122
54....	396.8136	397.9163	399.0204	400.1258	401.2326	402.3408	403.4448	404.5557	405.6679	406.7759
55....	407.8855	409.0017	410.1139	411.2273	412.3477	413.4639	414.5814	415.7002	416.8204	417.9419

* From Water-Supply and Irrigation Paper No. 180.

TABLE 58—Continued.

Head in feet.	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9
56....	419.0648	420.1833	421.3089	422.4257	423.5583	424.6879	425.8131	426.9453	428.0732	429.2080
57....	430.3386	431.4704	432.6036	433.7380	434.8738	436.0110	437.1494	438.2892	439.4302	440.5726
58....	441.7106	442.8556	443.9961	445.1438	446.2869	447.4372	448.5830	449.7300	450.8842	452.0359
59....	454.0849	455.3271	455.4907	456.6455	457.8017	458.9592	460.1179	461.2720	462.4334	463.5960
60....	464.7540	465.9192	467.0797	468.2475	469.4106	470.5750	471.7467	472.9137	474.0819	475.2514
61....	476.4222	477.3942	478.7676	479.9422	481.1181	482.2891	483.4676	484.6473	485.8222	487.0044
62....	488.1880	489.3666	490.5465	491.7339	492.9163	494.1000	495.2912	496.4774	497.6648	498.8536
63....	500.0436	501.2348	502.4273	503.6211	504.8161	506.0061	507.2036	508.4024	509.5961	510.7974
64....	512.0000	513.1974	514.3960	515.6024	516.8035	518.0059	519.2160	520.4209	521.6270	522.8344
65....	524.0430	525.2528	526.4639	527.6762	528.8898	530.1046	531.3120	532.5313	533.7498	534.9630
66....	536.1840	537.2996	538.6230	539.8411	541.0670	542.2875	543.5092	544.7389	545.9630	547.1884
67....	548.4151	549.6429	550.8720	552.1022	553.3337	554.5665	555.6179	557.0356	558.2652	559.5027
68....	560.7416	561.9748	563.2160	564.4516	565.6953	566.9334	568.1795	569.4199	570.6616	571.9113
69....	573.1554	574.4006	575.6473	576.8947	578.1436	579.3937	580.6449	581.8974	583.1510	584.4059
70....	585.6620	586.9122	588.1707	589.4303	590.6841	591.9462	593.2023	594.4668	595.7253	596.9921
71....	598.2531	599.5152	600.7856	602.0500	603.3157	604.5825	605.8505	607.1197	608.3901	609.6616
72....	610.9344	612.2083	613.4340	614.7596	616.0371	617.3085	618.5883	619.8692	621.0841	622.4274
73....	623.7120	624.9903	626.2699	627.5579	628.8398	630.1302	631.4144	632.6997	633.9862	635.2813
74....	636.5702	637.8602	639.1513	640.4437	641.7372	643.0318	644.3276	645.6246	646.9152	648.2145
75....	649.5150	650.8166	652.1118	653.4157	654.7208	656.0195	657.3268	658.6278	659.9375	661.2408
76....	662.5452	663.8583	665.1650	666.4728	667.7894	669.0996	670.4108	671.7131	673.0368	674.3514
77....	675.6673	676.9842	677.2043	679.6216	680.9419	682.2635	683.5784	684.9021	686.2271	687.5454
78....	688.8726	690.2009	691.5226	692.8532	694.1771	695.5100	696.8361	698.8361	699.1713	700.8292
79....	702.1599	703.4995	704.8324	706.1665	707.5016	708.8379	710.1752	711.5137	712.8534	714.1941
80....	715.5360	716.8789	718.2230	719.5683	720.9146	722.2540	723.6026	724.9523	726.2950	727.6496
81....	729.0000	730.3460	731.7613	733.0495	734.3989	735.7575	737.1091	738.4699	739.8237	741.1876
82....	742.5346	743.8998	745.2580	746.6173	747.9776	749.3392	750.7018	752.0655	753.4303	754.7962
83....	756.1632	757.5312	758.9094	760.2624	761.6338	763.0063	764.3798	765.7461	767.1219	768.4904
84....	769.8684	771.2474	772.6192	774.0004	775.3743	776.7493	778.1338	779.5110	780.8892	782.2770
85....	783.6575	785.0389	686.4215	787.8052	789.1984	790.5843	791.9712	793.3591	794.7482	796.1383
86....	797.5296	798.9219	800.3066	801.7011	803.0966	804.4932	805.8909	807.2810	808.6808	810.0833
87....	811.4751	812.8781	814.2736	815.6788	817.0763	818.4837	819.8834	821.2929	822.6947	824.1064
88....	825.5704	826.9154	828.3214	829.7374	831.1456	832.5549	833.9652	835.3766	836.7890	838.2025
89....	839.6171	841.0327	842.4494	843.8671	845.2859	846.7058	848.1267	849.5487	850.9627	852.3868
90....	853.8120	855.2382	856.6564	858.0847	859.5051	860.9355	862.3670	863.7905	865.2241	866.6496
91....	868.0763	869.5130	870.9417	872.3806	873.8114	875.2432	876.6761	878.1192	879.5541	880.9901
92....	882.4272	883.8652	885.3044	886.7445	888.1857	889.6280	891.0712	892.5156	893.9609	895.4073
93....	896.8548	898.3032	899.7528	901.1946	902.6456	904.0982	905.5519	906.9972	908.4530	909.9097
94....	911.3582	912.8170	914.2675	915.7284	917.1809	918.6439	920.0985	921.5541	923.0202	924.4778
95....	925.9365	927.4056	928.8664	930.3281	931.7908	933.2642	934.7290	936.1948	937.6616	939.1295
96....	940.5984	942.0683	943.5392	945.0111	946.4841	947.9581	949.4331	950.9091	952.3764	953.8545
97....	955.3336	956.8136	958.2948	959.7672	961.2503	962.7345	964.2099	965.6961	967.1735	968.6617
98....	970.1412	971.6314	973.1129	974.6051	976.0886	977.5829	979.0684	980.5548	982.0522	983.5407
99....	985.0302	986.5206	988.0220	989.5145	991.0080	992.5025	993.9980	995.4945	996.9920	998.4905
100...	1,000.0000	....	....	....	....	....	....	....	....	....

TABLE 59.

Table of five-halves ($\frac{5}{2}$) powers of number from 0 to 50 feet, varying by one-tenth foot.

Head in feet.	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9
0.....	0.000	.003	.018	.049	.101	.177	.279	.410	.572	.768
1.....	1.000	1.269	1.578	1.927	2.319	2.756	3.238	3.769	4.347	4.976
2.....	5.657	6.390	7.179	8.022	8.923	9.883	10.901	11.880	13.118	14.320
3.....	15.589	16.920	18.317	19.784	21.315	22.918	24.588	26.333	28.150	30.038
4.....	32.000	34.038	36.149	38.343	40.612	42.957	45.384	47.888	50.477	53.150
5.....	55.901	58.736	61.662	64.671	67.765	70.945	74.211	77.671	81.014	84.553
6.....	88.182	91.903	95.716	99.622	103.622	107.718	111.910	116.198	120.578	125.063
7.....	129.642	134.325	139.104	143.985	148.962	154.050	159.235	164.526	169.915	175.420
8.....	181.019	186.729	192.544	198.470	204.506	210.647	216.892	223.251	229.724	236.313
9.....	243.000	249.804	256.726	263.757	270.908	278.170	286.452	293.047	300.654	308.385
10.....	316.228	324.190	332.275	340.477	348.806	357.252	365.817	374.511	383.314	392.258
11.....	401.311	410.500	419.798	429.242	438.797	448.477	458.293	468.234	478.301	488.507
12.....	498.820	509.301	519.879	530.610	541.446	552.438	563.548	574.802	586.163	597.696
13.....	609.336	621.137	633.046	645.170	657.297	669.641	682.094	694.727	707.457	720.354
14.....	733.365	746.539	759.842	773.301	786.873	800.618	814.476	828.521	842.668	856.988
15.....	871.416	886.038	900.767	915.659	930.684	945.872	961.194	976.697	992.303	1008.032
16.....	1024.000	1040.092	1056.305	1072.703	1089.206	1105.896	1122.724	1139.708	1156.831	1174.144
17.....	1191.578	1209.192	1226.945	1244.856	1262.909	1281.140	1299.514	1318.066	1336.744	1355.621
18.....	1374.606	1393.809	1413.121	1432.634	1452.257	1472.082	1492.055	1512.194	1532.482	1552.956
19.....	1573.561	1594.373	1615.296	1636.428	1657.691	1679.145	1700.751	1722.529	1744.459	1766.583
20.....	1788.840	1811.312	1833.918	1856.719	1879.636	1902.769	1926.059	1949.526	1973.130	1996.953
21.....	2020.914	2045.075	2069.374	2093.875	2118.536	2143.378	2168.381	2193.588	2218.935	2244.465
22.....	2270.136	2296.057	2322.142	2348.368	2374.758	2401.380	2428.121	2455.096	2482.213	2509.519
23.....	2536.992	2564.678	2592.507	2620.551	2648.740	2677.167	2705.716	2734.482	2763.394	2792.524
24.....	2821.800	2851.343	2881.010	2910.848	2940.859	2971.115	3001.495	3032.123	3062.874	3093.875
25.....	3125.000	3156.375	3187.876	3219.627	3251.505	3283.661	3315.942	3348.402	3381.038	3413.905
26.....	3446.924	3480.200	3513.603	3547.239	3581.054	3615.077	3649.254	3683.666	3718.232	3753.034
27.....	3787.992	3823.187	3858.538	3894.127	3929.872	3965.830	4001.945	4038.328	4074.868	4111.623
28.....	4148.536	4185.692	4223.006	4260.565	4298.283	4336.247	4374.370	4412.711	4451.242	4489.991
29.....	4528.930	4568.089	4607.410	4646.980	4686.713	4726.697	4766.843	4807.212	4847.745	4888.530
30.....	4929.510	4970.714	5012.052	5053.676	5095.466	5137.512	5179.693	5222.131	5264.736	5307.600
31.....	5350.631	5393.891	5437.349	5481.037	5524.893	5569.011	5613.298	5657.816	5702.535	5747.487
32.....	5792.608	5837.995	5883.552	5929.376	5975.338	6021.568	6067.968	6114.638	6161.480	6208.559
33.....	6255.810	6303.365	6351.060	6398.995	6447.135	6495.516	6544.070	6592.899	6641.903	6691.148
34.....	6740.568	6790.879	6841.368	6890.904	6940.613	6991.149	7041.896	7092.923	7144.092	7195.542
35.....	7247.170	7299.080	7351.168	7403.504	7456.019	7508.960	7562.081	7615.167	7668.432	7722.126
36.....	7776.000	7830.133	7884.447	7939.028	7993.789	8051.024	8104.060	8159.555	8215.232	8271.179
37.....	8327.309	8383.709	8440.293	8497.149	8554.188	8611.515	8669.026	8726.796	8784.750	8842.995
38.....	8901.424	8960.114	9018.989	9078.157	9137.510	9197.142	9256.959	9317.056	9377.339	9437.902
39.....	9498.653	9559.684	9620.903	9682.388	9744.061	9806.083	9868.193	9930.637	9993.271	10056.189
40.....	10119.296	10182.673	10246.240	10310.110	10374.171	10438.519	10503.058	10567.869	10632.872	10698.164
41.....	10763.648	10829.423	10895.389	10961.648	11028.099	11094.842	11161.779	11228.993	11296.340	11364.118
42.....	11432.030	11502.221	11568.607	11637.288	11706.165	11775.356	11844.743	11939.996	11984.205	12054.351
43.....	12124.693	12195.335	12266.173	12337.313	12408.650	12480.272	12552.091	12624.213	12696.634	12769.197
44.....	12841.761	12915.113	12988.340	13062.014	13135.829	13209.950	13284.271	13358.881	13433.692	13508.794
45.....	13584.096	13659.726	13735.557	13811.680	13888.005	13964.623	14041.444	14118.576	14195.912	14273.560
46.....	14351.411	14429.558	14507.909	14586.574	14665.444	14744.578	14823.982	14899.897	14976.000	15059.965
47.....	15144.152	15224.849	15305.752	15386.974	15468.402	15550.151	15632.107	15714.364	15796.829	15879.597
48.....	15962.573	16045.809	16129.325	16203.457	16297.190	16381.302	16465.928	16552.036	16635.783	16722.212
49.....	16807.000	16892.786	16978.851	17065.195	17152.070	17238.979	17326.168	17413.647	17501.393	17589.181
50....	17677.500	....	....	....	....	....	....	....	....	....

TABLE 60.

Velocities, in feet per second, due to heads from 0 to 50 feet.

Head in feet.	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9
0.....	0.000	2.536	3.587	4.393	5.072	5.671	6.212	6.710	7.173	7.609
1.....	8.020	8.412	8.786	9.144	9.490	9.823	10.145	10.457	10.760	11.055
2.....	11.342	11.622	11.896	12.163	12.425	12.681	12.932	13.179	13.420	13.658
3.....	13.891	14.121	14.347	14.569	14.789	15.004	15.217	15.427	15.634	15.839
4.....	16.040	16.240	16.437	16.631	16.823	17.013	17.201	17.387	17.571	17.753
5.....	17.934	18.112	18.289	18.464	18.637	18.809	18.979	19.148	19.315	19.481
6.....	19.645	19.808	19.970	20.131	20.290	20.448	20.604	20.760	20.914	21.067
7.....	21.219	21.370	21.520	21.669	21.817	21.964	22.110	22.255	22.399	22.542
8.....	22.685	22.826	22.966	23.106	23.245	23.383	23.520	23.656	23.792	23.927
9.....	24.061	24.194	24.326	24.458	24.589	24.720	24.850	24.979	25.107	25.235
10.....	25.362	25.489	25.614	25.740	25.864	25.988	26.112	26.235	26.357	26.479
11.....	26.600	26.721	26.841	26.960	27.079	27.198	27.316	27.433	27.550	27.667
12.....	27.783	27.898	28.013	28.128	28.242	28.356	28.469	28.582	28.694	28.806
13.....	28.917	29.028	29.139	29.249	29.359	29.468	29.577	29.686	29.794	29.901
14.....	30.009	30.116	30.222	30.329	30.435	30.540	30.645	30.750	30.854	30.958
15.....	31.062	31.165	31.268	31.371	31.474	31.576	31.677	31.779	31.880	31.980
16.....	32.081	32.181	32.281	32.380	32.480	32.579	32.677	32.775	32.873	32.971
17.....	33.068	33.165	33.262	33.359	33.455	33.551	33.647	33.742	33.837	33.932
18.....	34.027	34.121	34.215	34.309	34.403	34.496	34.589	34.682	34.775	34.867
19.....	34.959	35.051	35.143	35.234	35.325	35.416	35.507	35.597	35.688	35.778
20.....	35.867	35.957	36.046	36.135	36.224	36.313	36.401	36.490	36.578	36.666
21.....	36.753	36.841	36.928	37.015	37.102	37.188	37.275	37.361	37.447	37.532
22.....	37.618	37.703	37.789	37.874	37.959	38.043	38.128	38.212	38.296	38.380
23.....	38.464	38.547	38.630	38.714	38.797	38.879	38.962	39.044	39.127	39.209
24.....	39.291	39.373	39.454	39.536	39.617	39.698	39.779	39.860	39.940	40.021
25.....	40.101	40.181	40.261	40.341	40.421	40.500	40.579	40.659	40.738	40.816
26.....	40.895	40.974	41.052	41.130	41.209	41.287	41.364	41.442	41.520	41.597
27.....	41.674	41.751	41.828	41.905	41.982	42.058	42.135	42.211	42.287	42.363
28.....	42.439	42.515	42.590	42.666	42.741	42.816	42.891	42.966	43.041	43.116
29.....	43.190	43.264	43.339	43.413	43.487	43.561	43.635	43.708	43.782	43.855
30.....	43.928	44.002	44.075	44.148	44.220	44.293	44.366	44.438	44.510	44.582
31.....	44.655	44.727	44.798	44.870	44.942	45.013	45.085	45.156	45.227	45.298
32.....	45.369	45.440	45.511	45.581	45.652	45.722	45.792	45.863	45.933	46.003
33.....	46.073	46.142	46.212	46.281	46.351	46.420	46.489	46.559	46.628	46.697
34.....	46.765	46.834	46.903	46.971	47.040	47.108	47.176	47.244	47.312	47.380
35.....	47.448	47.516	47.584	47.651	47.719	47.786	47.853	47.920	47.987	48.054
36.....	48.121	48.188	48.255	48.321	48.388	48.454	48.521	48.487	48.653	48.719
37.....	48.785	48.851	48.917	48.982	49.048	49.113	49.179	49.244	49.310	49.375
38.....	49.440	49.505	49.570	49.635	49.699	49.764	49.829	49.893	49.958	50.022
39.....	50.086	50.150	50.214	50.278	50.342	50.406	50.470	50.534	50.597	50.661
40.....	50.724	50.788	50.851	50.914	50.977	51.040	51.103	51.166	51.229	51.292
41.....	51.354	51.417	51.479	51.542	51.604	51.667	51.729	51.791	51.853	51.915
42.....	51.977	52.039	52.100	52.162	52.224	52.285	52.347	52.408	52.470	52.531
43.....	52.592	52.653	52.714	52.775	52.836	52.897	52.958	53.018	53.079	53.139
44.....	53.200	53.260	53.321	53.381	53.441	53.501	53.561	53.621	53.681	53.741
45.....	53.801	53.861	53.921	53.980	54.040	54.099	54.159	54.218	54.277	54.336
46.....	54.396	54.455	54.514	54.573	54.632	54.690	54.749	54.808	54.867	54.925
47.....	54.984	55.042	55.101	55.159	55.217	55.275	55.334	55.392	55.450	55.508
48.....	55.566	55.623	55.681	55.739	55.797	55.854	55.912	55.969	56.027	56.084
49.....	56.141	56.199	56.256	56.313	56.370	56.427	56.484	56.541	56.598	56.655

APPENDIX B

TEST DATA OF TURBINE WATER WHEELS.

TABLE 61.

Test of a 113-inch Boott Center Vent Turbine. Built in 1849 for the Boott Cotton Mills, Lowell, Mass., after designs by James B. Francis.

Number of experiment.	Gate opening (proportional part).	Proportional discharge (discharge at full gate with highest efficiency=1).	Mean head in feet.	Duration of test in minutes.	Revolutions per minute.	Discharge in second-feet.	Horsepower developed.	Percentage of efficiency.
1	2	3	4	5	6	7	8	9
1........	0.25	0.600	14.60	4	35.6	67.53	42.2	37.7
2........	0.25	0.577	14.67	7	52.7	64.89	21.94	20.3
3........	0.25	0.590	14.57	8	43.4	66.43	36.4	33.2
4........	0.25	0.592	14.16	17	32.7	66.61	40.8	38.2
5........	0.25	0.596	14.20	12	30.0	67.03	41.1	38.1
6........	0.25	0.595	14.14	7	27.0	66.89	40.0	37.3
7........	0.25	0.598	14.24	7	25.25	67.37	29.3	27.0
8........	0.25	0.544	14.30	8	60.8	61.08	0.0	0.0
9........	0.50	0.756	14.29	11	60.6	85.0	41.7	30.3
10........	0.50	0.767	14.23	10	57.6	86.35	52.4	37.6
11........	0.50	0.775	14.20	10	55.9	87.08	57.8	41.3
12........	0.50	0.780	14.19	8	54.1	87.69	62.7	44.4
13........	0.50	0.785	14.19	14	51.4	88.29	69.5	48.9
14........	0.50	0.802	13.78	11	41.6	90.17	82.0	58.3
15........	0.50	0.815	13.61	11	35.9	91.70	84.4	59.6
16........	0.50	0.685	13.95	8	70.6	77.11	0.0	0.0
17........	0.75	0.852	13.52	7	66.3	95.76	35.9	24.5
18........	0.75	0.876	13.37	2.5	59.6	98.49	65.3	43.7
19........	0.75	0.893	13.37	7	54.1	100.42	87.6	57.6
20........	0.75	0.910	13.40	9	49.5	102.42	102.2	65.7
21........	0.75	0.913	13.38	12	47.2	102.82	107.7	69.0
22........	0.75	0.920	13.34	9	44.8	103.52	111.2	71.0
23........	0.75	0.922	13.32	9	43.5	103.77	112.8	72.0
24........	0.75	0.921	13.33	9	42.6	103.69	114.0	72.8
25........	0.75	0.925	13.30	10	41.9	104.23	114.9	73.1
26........	0.75	0.818	13.70	8	75.2	92.02	0.0	0.0
27........	1.00	1.000	13.40	9	42.5	112.53	136.4	79.7
28........	1.00	1.005	13.43	6	41.9	112.99	137.0	79.6
29........	1.00	1.000	13.33	6	40.7	112.56	135.6	79.7
30........	1.00	1.005	13.38	7	40.3	113.00	136.6	79.7
31........	1.00	1.006	13.39	8	39.6	113.07	136.8	79.7
32........	1.00	1.007	13.38	8	38.9	113.16	136.9	79.7
33........	1.00	1.006	13.36	9	38.1	113.09	136.6	79.8
34........	1.00	1.010	13.38	8	37.4	113.67	136.5	79.2
35........	1.00	1.017	13.40	8	36.8	114.29	136.7	78.7
36........	1.00	1.013	13.32	8	35.5	113.97	134.5	78.1
37........	1.00	0.982	13.54	4	0.0	110.45	0.0	0.0
38........	1.00	0.980	13.57	2	0.0	110.32	0.0	0.0
39........	1.00	0.887	13.60	7	77.0	99.8	0.0	0.0

TABLE 62.

Test of an 81-inch Fourneyron Turbine Built in 1851 for the Tremont Mills, Lowell, Mass., after designs by James B. Francis.

Number of experiment.	Gate opening (proportional part).	Proportional discharge (discharge at full gate with highest efficiency=1).	Mean head in feet.	Duration of test in minutes.	Revolutions per minute.	Discharge in second-feet.	Horsepower developed.	Percentage of efficiency.
1	2	3	4	5	6	7	8	9
1........	1.0	1.01	12.86	8	53.62	139.42	161.4	78.4
2........	1.0	1.01	12.86	10	53.5	139.42	159.2	78.3
3........	1.0	1.01	12.87	10	53.6	139.47	159.5	78.4
4........	1.0	1.13	12.55	6	95.8	156.65	60.6	27.2
5........	1.0	1.12	12.61	8	91.9	154.39	77.9	35.3
6........	1.0	1.10	12.65	7	87.7	152.27	94.0	43.0
7........	1.0	1.08	12.70	7	83.0	149.46	109.2	50.7
8........	1.0	1.07	12.72	5	78.5	147.29	121.5	57.2
9........	1.0	1.06	12.78	5	77.4	146.02	131.7	62.2
10........	1.0	1.05	12.80	8	71.0	144.87	140.2	66.7
11........	1.0	1.04	12.82	9	67.5	143.91	147.2	70.3
12........	1.0	....	....	9	107.0	....	0.0	0.0
13........	1.0	1.18	12.51	9	107.0	163.43	0.0	0.0
14........	1.0	1.03	12.86	9	64.0	142.52	152.6	73.5
15........	1.0	1.03	12.89	9	61.4	142.04	155.8	75.0
16........	1.0	1.03	12.89	8	60.0	141.98	157.0	75.6
17........	1.0	1.02	12.90	9	58.2	141.28	158.3	76.6
18........	1.0	1.02	12.88	7	56.7	140.47	158.9	77.4
19........	1.0	1.01	12.88	10	55.4	140.08	159.5	77.9
20........	1.0	1.01	12.87	9	54.7	140.01	159.7	78.0
21........	1.0	1.01	12.90	10	54.1	139.90	160.0	78.1
22........	1.0	1.01	12.90	14	53.8	139.67	160.2	78.4
23........	1.0	1.17	12.43	9	106.8	161.69	0.0	0.0
24........	1.0	1.01	12.90	9	53.6	139.01	160.5	78.9
25........	1.0	1.01	12.90	13	53.1	139.03	160.4	78.8
26........	1.0	1.0	12.89	5	52.5	138.76	159.5	78.6
27........	1.0	1.0	12.90	14	52.8	138.85	160.5	79.0
28........	1.0	1.0	12.91	13	52.4	138.87	160.5	78.9
29........	1.0	1.0	12.91	12	52.0	138.51	160.6	79.2
30........	1.0	1.0	12.90	12	51.1	138.19	160.5	79.4
31........	1.0	1.17	12.54	9	106.8	162.32	0.0	0.0
32........	1.0	1.0	12.91	6	50.2	138.27	160.6	79.3
33........	1.0	1.0	12.93	10	48.8	138.23	160.6	79.2
34........	1.0	1.0	12.94	11	47.1	138.09	160.0	78.9
35........	1.0	1.0	12.94	11	44.5	137.71	158.4	78.3
36........	1.0	0.99	12.96	11	41.7	136.49	156.2	77.9
37........	1.0	0.98	12.94	10	38.7	135.14	152.6	77.0
38........	1.0	1.17	12.5	9	107.1	161.69	0.0	0.0
39........	1.0	0.98	12.96	12	38.8	135.34	153.0	76.9
40........	1.0	0.97	12.97	8	36.0	134.80	149.3	75.3
41........	1.0	0.97	12.98	11	31.9	133.75	142.7	72.5
42........	1.0	0.97	12.95	9	27.3	133.43	133.0	67.9
43........	1.0	0.98	12.80	1.5	0.0	135.65	0.0	0.0
44........	1.0	0.98	12.77	2.5	0.0	135.62	0.0	0.0
45........	1.0	1.17	12.47	9	106.9	162.02	0.0	0.0
46........	1.0	1.00	12.95	11	49.9	138.62	161.1	79.1
47........	1.0	1.00	12.93	10	49.0	138.50	160.7	79.1
48........	1.0	1.00	12.95	11	48.2	138.47	160.5	78.9
49........	1.0	1.00	12.95	12	47.4	138.37	160.3	78.9
50........	1.0	1.00	12.95	11	46.2	138.16	159.8	78.7
51........	0.75	1.04	12.76	7	89.0	143.33	71.7	34.6
52........	0.75	1.01	12.87	8	75.4	139.21	120.6	59.4
53........	0.75	1.00	12.91	9	68.5	137.75	136.1	67.5
54........	0.75	1.00	12.94	8	64.7	137.80	142.8	70.6
55........	0.75	0.99	12.95	8	61.4	137.00	145.9	72.5
56........	0.75	0.98	12.95	8	57.9	135.54	148.3	74.5
57........	0.75	0.98	12.96	8	55.8	135.10	149.0	75.0
58........	0.75	0.97	12.98	8	54.0	134.33	149.8	75.8
59........	0.75	0.96	13.00	9	51.9	133.30	149.5	76.0

TABLE 62—Continued.

1	2	3	4	5	6	7	8	9
60........	0.75	0.96	13.01	9	50.1	132.73	149.3	76.2
61........	0.75	0.95	13.03	9	48.3	132.00	148.7	76.3
62........	0.75	0.95	13.04	9	46.3	130.99	147.9	76.4
63........	0.75	0.95	13.03	8	42.5	130.89	145.1	75.0
64........	0.75	1.08	12.72	11	103.0	149.55	0.0	0.0
65........	0.49	0.88	13.17	11	92.9	121.97	0.0	0.0
66........	0.49	0.86	13.08	6	81.1	118.55	52.8	30.0
67........	0.49	0.84	13.13	6	73.1	116.1	78.3	45.3
68........	0.49	0.83	13.18	8	65.0	114.26	96.6	56.6
69........	0.49	0.82	13.21	7	60.2	113.24	103.3	60.9
70........	0.49	0.81	13.25	6	55.4	111.52	106.7	63.7
71........	0.49	0.79	13.28	7	50.6	109.71	107.8	65.2
72........	0.49	0.78	13.31	9	46.5	108.05	106.8	65.5
73........	0.49	0.78	13.31	6	46.5	107.95	107.0	65.6
74........	0.49	0.76	13.33	9	41.2	105.53	102.3	64.1
75........	0.49	0.75	13.36	8	36.9	103.85	97.3	61.9
76........	0.49	0.73	13.41	7	27.4	100.54	83.7	54.8
77........	0.87	0.99	12.88	7	51.3	137.36	156.8	78.1
78........	0.87	0.99	12.90	7	49.3	136.97	157.0	78.4
79........	0.87	0.99	12.91	7	47.4	136.55	156.6	78.3
80........	0.25	0.58	13.35	5	74.9	80.45	0.0	0.0
81........	0.25	0.57	13.37	6	68.8	78.84	168.2	14.1
82........	0.25	0.56	13.40	6	57.6	76.62	38.6	33.2
83........	0.25	0.54	13.43	6	46.3	74.06	49.7	44.0
84........	0.25	0.52	13.48	8	40.3	71.87	50.9	46.3
85........	0.25	0.51	13.51	8	33.6	70.01	48.8	45.5
86........	0.25	0.49	13.56	7	27.7	67.82	44.4	42.7
87........	0.25	0.47	13.56	11	18.0	64.51	32.7	33.0
88........	0.25	0.44	13.52	....	0.0	60.36	0.0	0.0
89........	0.25	0.44	13.53	....	0.0	60.42	0.0	0.0
90........	0.087	0.28	13.98	7	37.2	38.22	9.09	15.0
91........	0.087	0.28	14.00	7	41.3	38.57	6.23	10.2
92........	0.087	0.27	14.02	17	23.3	37.17	14.21	24.0

TABLE 63.

Holyoke Test of a 30-inch Special Chase Jonval Turbine. Test No. 256. June 7, 1884.

Number of experiment.	Gate opening (proportional part).	Proportional discharge (discharge at full gate with highest efficiency=1).	Mean head in feet.	Duration of test in minutes.	Revolutions per minute.	Discharge in second-feet.	Horse-power developed.	Percentage of efficiency.
1	2	3	4	5	6	7	8	9
1........	1.000	0.960	14.72	3	Still.	39.74	....	....
8........	1.000	1.008	14.51	3	169.67	41.42	49.66	72.98
7........	1.000	1.004	14.43	3	181.67	41.16	50.68	75.36
6........	1.000	1.004	14.61	3	196.33	41.42	52.08	76.00
5........	1.000	0.998	14.57	4	204.75	41.10	51.50	75.96
4........	1.000	0.999	14.41	3	215.00	40.91	51.13	76.60
3........	1.000	1.001	14.41	3	225.00	41.02	50.42	75.34
2........	1.000	0.998	14.49	5	244.80	40.99	50.38	74.91
22........	0.930	0.922	14.68	4	185.75	38.12	49.27	77.76
21........	0.930	0.920	14.72	6	194.17	38.07	49.73	78.37
23........	0.930	0.919	14.78	4	203.75	38.12	50.32	78.88
24........	0.930	0.916	14.87	4	213.00	38.13	50.66	78.90
25........	0.930	0.913	14.95	4	226.00	38.07	50.65	78.59
17........	0.837	0.831	15.28	3	180.00	35.07	46.10	75.98
16........	0.837	0.827	15.42	4	191.00	35.03	47.17	77.12
15........	0.837	0.826	15.42	3	199.67	35.00	47.49	77.71
18........	0.837	0.823	15.33	3	207.00	34.75	47.34	78.47
19........	0.837	0.822	15.28	4	215.00	34.65	47.20	78.73
20........	0.837	0.818	15.26	4	228.25	34.49	46.98	78.82
14........	0.674	0.666	16.13	4	163.50	28.85	35.15	66.70
13........	0.674	0.663	16.14	4	174.25	28.75	35.86	68.25
12........	0.674	0.661	16.07	3	185.33	28.60	36.45	70.04
11........	0.674	0.669	16.11	4	195.50	28.52	36.66	70.46
10........	0.674	0.655	16.14	4	206.50	28.40	36.83	70.97
9........	0.674	0.650	16.20	4	217.00	28.25	36.72	70.86
30........	0.488	0.462	17.10	4	142.25	20.62	16.26	40.74
29........	0.488	0.460	17.11	4	158.50	20.55	16.67	41.88
28........	0.488	0.458	17.08	3	174.33	20.43	16.74	42.38
27........	0.488	0.459	17.02	4	182.50	20.43	16.69	42.40
26........	0.488	0.458	17.07	3	190.67	20.43	16.57	41.96
31........	0.488	0.457	17.09	5	206.20	20.37	16.03	40.67

TABLE 64.

Holyoke Test of a 30-inch Regular Chase Jonval Turbine. June 10, 1884.

Number of experiment.	Gate opening (proportional part).	Proportional discharge (discharge at full gate with highest efficiency=1).	Mean head in feet.	Duration of test in minutes.	Revolutions per minute.	Discharge in second-feet.	Horsepower developed.	Percentage of efficiency.
1	2	3	4	5	6	7	8	9
1........	1.000	0.938	15.62	3	Still.	33.33	....	....
6........	1.000	0.993	15.32	3	194.00	34.93	43.48	71.70
5........	1.000	0.995	15.29	3	201.67	34.96	43.35	71.58
4........	1.000	0.996	15.30	5	211.14	35.00	43.46	71.62
3........	1.000	1.001	15.27	3	222.33	35.14	43.72	71.92
2........	1.000	1.007	15.26	3	237.00	35.34	43.36	70.96
29........	0.889	0.894	15.78	4	174.75	31.93	41.56	72.80
28........	0.889	0.897	15.77	4	190.25	32.01	42.64	74.55
27........	0.889	0.897	15.75	4	200.25	32.01	43.05	75.36
26........	0.889	0.898	15.80	4	211.25	32.10	43.48	75.66
23........	0.889	0.901	15.77	3	220.67	32.17	43.40	75.50
24........	0.889	0.903	15.74	4	232.00	32.20	43.50	75.76
25........	0.889	0.907	15.72	3	242.67	32.33	43.29	75.17
22........	0.733	0.756	16.30	4	184.50	27.43	36.28	71.63
21........	0.733	0.757	16.24	4	195.25	27.41	36.61	72.59
20........	9.733	0.758	16.27	4	207.00	27.47	36.92	72.91
18........	0.733	0.756	16.28	3	218.67	27.43	37.00	73.14
19........	0.733	0.757	16.32	4	230.50	27.47	36.90	72.64
14........	0.611	0.644	16.65	5	175.80	23.63	27.34	61.33
13........	0.611	0.644	16.65	4	188.75	23.60	27.62	62.05
12........	0.611	0.641	16.68	3	202.00	23.54	27.72	62.16
15........	0.611	0.644	16.77	3	209.33	23.63	27.76	62.21
16........	0.611	0.644	16.67	3	221.33	23.64	27.33	61.21
17........	0.611	0.647	16.61	3	236.33	23.71	27.02	60.56
7........	0.411	0.469	17.14	3	141.33	17.47	12.93	38.10
8........	0.411	0.469	17.20	4	157.00	17.47	12.92	37.96
9........	0.411	0.468	17.17	4	166.00	17.43	12.91	38.06
10........	0.411	0.469	17.13	4	184.00	17.43	12.62	36.46
11........	0.411	0.471	17.16	3	200.00	17.53	11.89	34.89

TABLE 65.

Experiments Made for the Tremont and Suffolk Mills at Holyoke Testing Flume, December 3–5, 1890, on a 48-inch Victor Turbine, with Cylinder Gate.

	Height of speed-gate opening. Inches.		H. Fall at the wheel. Feet.	Duration of the experiment. Seconds.	Revolutions per minute.	Quantity of water discharged by the wheel. Cubic feet per second.	Horse-power developed.	Percentage of efficiency.
1	2	3	4	5	6	7	8	9
	Full Gate.							
....	20.58	......	13.185	360.	142.7	94.49	....	....
....	20.58	......	.019	300.	121.8	103.75	74.07	43.65
....	20.58	......	12.951	300.	104.6	115.87	127.22	74.75
....	20.58	......	13.200	360.	100.3	119.44	140.33	78.49
....	20.58	......	.167	300.	96.0	121.18	145.95	80.66
....	20.58	......	.262	360.	93.6	122.67	150.81	81.74
....	20.58	......	.265	360.	91.2	123.41	152.60	82.20
....	20.58	......	.157	360.	87.8	123.67	152.23	82.49
....	20.58	......	.139	180.	84.3	124.05	151.29	81.84
....	20.58	......	.140	360.	83.0	124.17	151.42	81.83
....	20.58	......	.157	420.	81.6	124.55	151.43	81.48
....	20.58	......	.130	360.	79.8	124.67	150.50	81.07
....	20.58	......	.095	540.	77.3	124.93	148.97	80.29
....	20.58	......	.221	300.	75.0	126.05	150.51	79.64
....	20.58	......	.201	300.	70.1	126.67	147.07	77.55
....	20.58	......	.081	420.	Still	129.60	....	....
	¾ Gate (nearly).							
....	17.90	......	13.463	300.	108.2	108.38	115.15	69.59
....	17.90	......	.389	300.	103.2	110.90	125.52	74.54
....	17.90	......	.324	360.	97.7	112.71	133.64	78.46
....	17.90	......	.252	360.	91.6	114.65	139.24	80.81
....	17.90	......	.235	360.	89.2	115.26	140.98	81.49
....	17.90	......	.195	300.	86.2	115.87	141.53	81.63
....	17.90	......	.169	480.	83.2	116.36	141.75	81.57
....	17.90	......	.242	360.	77.3	117.72	141.08	79.80
....	17.90	......	.199	240.	70.1	118.34	136.46	77.04
....	17.90	......	.161	300.	Still	118.70	....	....
....	17.90	......	14.023	300.	145.1	93.35	....	....
	⅞ Gate (nearly).							
....	15.38	......	13.043	300.	102.0	98.85	105.45	72.12
....	15.38	......	12.985	300.	95.2	101.53	115.79	77.44
....	15.38	......	13.049	240.	90.5	103.28	121.08	79.22
....	15.38	......	.049	300.	86.5	104.70	123.62	79.78
....	15.38	......	.029	300.	84.0	105.29	125.15	80.44
....	15.38	......	.029	300.	81.0	106.00	125.61	80.20
....	15.38	......	.017	300.	77.5	106.59	124.89	79.37
....	15.38	......	.025	300.	73.0	107.18	122.08	77.11
....	15.38	......	.024	180.	68.8	107.42	119.30	75.19
	⅝ Gate (nearly).							
....	12.90	......	13.193	300.	103.2	87.40	87.86	67.19
....	12.90	......	.159	360.	98.0	89.19	95.35	71.64
....	12.90	......	.110	180.	93.3	90.30	99.33	73.98
....	12.90	......	.165	300.	91.3	91.54	102.72	75.15
....	12.90	......	.139	360.	88.1	92.33	104.45	75.92
....	12.90	......	.097	180.	84.3	93.12	105.13	76.01
....	12.90	......	.062	240.	79.2	94.15	106.03	76.02
....	12.90	......	12.993	300.	73.1	94.94	104.47	74.67
	½ Gate (nearly).							
....	10.34	......	13.239	300.	101.4	74.98	70.91	62.99
....	10.34	......	.195	300.	95.9	76.25	75.81	66.44
....	10.34	......	.137	360.	90.0	77.52	79.36	68.71
....	10.34	......	.113	300.	86.2	78.16	81.25	69.90
....	10.34	......	.063	300.	82.6	78.91	82.88	70.90
....	10.34	......	.027	360.	78.5	79.66	83.54	70.99
....	10.34	......	.010	180.	74.0	80.52	83.25	70.08
....	10.34	......	12.999	300.	72.0	80.73	83.19	69.90
....	10.34	......	13.254	240.	Still	79.33	....	....
....	10.34	......	.437	300.	129.1	65.62	....	....

TABLE 65—Continued.

1	2		3	4	5	6	7	8	9
....	⅜ Gate (nearly).	7.80	......	13.215	300.	102.5	59.59	43.63	48.86
....		7.80	......	.200	360.	98.4	60.27	47.88	53.07
....		7.80	......	.147	360.	90.7	61.45	52.43	57.22
....		7.80	......	.084	360.	86.2	62.04	55.02	59.77
....		7.80	......	.039	360.	81.3	62.73	56.88	61.32
....		7.80	......	.007	360.	76.2	63.43	57.90	61.88
....		7.80	......	12.967	300.	71.5	64.02	58.70	62.35
....	¼ Gate (nearly).	5.23	......	13.094	240.	102.5	42.65	15.58	24.60
....		5.23	....	.059	300.	97.0	43.18	20.65	32.28
....		5.23	......	.019	300.	91.3	43.79	24.98	38.64
....		5.23	......	.005	360.	88.1	43.96	26.78	41.31
....		5.23	......	12.972	360.	85.2	44.23	28.49	43.78
....		5.23	......	.939	240.	81.9	44.40	29.87	45.85
....		5.23	......	13.213	300.	80.8	45.19	31.94	47.17
....		5.23	......	.175	360.	74.3	45.73	33.90	49.62
....		5.23	......	.133	300.	67.6	46.17	34.94	50.81

TABLE 66.

Holyoke Test of a 36-inch Right Hand Swain Turbine. Test No. 977. Date Jan. 20–21, 1897.

Number of experiment.	Gate opening (proportional part).	Proportional discharge (discharge at full gate with highest efficiency=1).	Mean head in feet.	Duration of test in minutes.	Revolutions per minute.	Discharge in second-feet.	Horsepower developed.	Percentage of efficiency.
1	2	3	4	5	6	7	8	9
64........	1.000	1.004	15.16	3	130.33	76.49	111.57	84.84
63........	1.000	.995	15.25	4	135.00	75.98	111.47	84.83
62........	1.000	.984	15.40	3	140.33	75.52	111.61	84.62
61........	1.000	.973	15.42	3	144.00	74.74	110.16	84.28
60........	1.000	.966	15.43	4	146.50	74.19	108.51	83.58
59........	1.000	.954	15.48	4	150.75	73.45	107.08	83.04
58........	1.000	.945	15.47	4	154.00	72.73	104.72	82.07
57........	1.000	.934	15.44	3	158.00	71.76	101.68	80.92
56........	1.000	.922	15.33	4	161.37	70.58	97.97	79.84
55........	.875	.932	15.16	3	132.00	70.95	102.58	84.10
54........	.875	.925	15.15	4	135.75	70.42	102.20	84.47
53........	.875	.916	15.28	4	140.75	70.01	102.54	84.52
52........	.875	.907	15.41	4	147.00	69.61	102.63	84.37
51........	.875	.896	15.50	4	153.50	68.99	101.58	83.75
50........	.875	.877	15.60	3	162.33	67.75	98.55	82.22
49........	.750	.866	15.66	4	130.00	67.03	100.24	84.20
48........	.750	.857	15.74	4	136.00	66.52	100.73	84.83
47........	.750	.849	15.76	4	141.00	65.96	100.16	84.96
46........	.750	.844	15.70	2	146.00	65.41	99.28	85.24
45........	.750	.838	15.54	4	149.75	64.64	97.28	85.39
44........	.750	.829	15.62	3	157.33	64.09	96.47	84.97
43........	.750	.826	15.16	4	156.76	62.88	91.36	84.51
42........	.750	.814	15.20	4	162.75	62.06	88.93	83.13
41........	.625	.783	15.30	4	127.50	59.90	85.92	82.67
40........	.625	.775	15.38	4	134.75	59.45	86.72	83.63
39........	.625	.767	15.43	4	142.12	58.95	87.15	84.48
38........	.625	.758	15.47	4	149.50	58.30	86.23	84.30
37........	.625	.749	15.51	4	154.50	57.70	84.42	83.18
36........	.625	.733	15.58	4	162.75	56.62	81.02	80.99
35........	.625	.715	15.65	4	169.50	55.32	76.15	77.56
34........	.500	.683	15.74	5	123.20	52.98	74.80	79.09
33........	.500	.676	15.78	4	131.25	52.54	75.70	80.51
32........	.500	.668	15.83	4	137.50	51.98	75.13	80.51
31........	.500	.660	15.85	4	144.00	51.42	74.31	80.40
30........	.500	.652	15.91	4	150.25	50.87	72.98	79.51
29........	.500	.644	15.95	4	157.50	50.31	71.72	78.81
28........	.500	.635	15.96	3	163.33	49.62	69.41	77.29
26........	.375	.557	15.21	4	120.37	42.50	54.08	73.77
25........	.375	.552	15.19	4	127.50	42.06	54.19	74.78
24........	.375	.545	15.21	4	123.50	41.58	53.49	74.58
23........	.375	.538	15.21	4	139.00	41.06	52.32	73.87
22........	.375	.531	15.22	3	145.67	40.53	51.30	73.32
21........	.375	.524	15.21	3	152.33	40.01	49.94	72.36
20........	.375	.514	15.30	4	160.25	39.36	48.65	71.23
27........	.375	.504	15.30	5	167.20	38.57	45.68	68.25
19........	.250	.420	15.53	4	113.50	32.40	36.52	64.00
18........	.250	.416	15.54	4	120.25	32.11	36.50	64.50
17........	.250	.412	15.57	4	126.50	31.83	36.10	64.22
16........	.250	.407	15.61	4	133.25	31.46	35.60	63.91
15........	.250	.401	15.60	4	139.50	31.00	34.72	63.31
14........	.250	.396	15.61	4	146.25	30.59	33.74	62.31
13........	.250	.386	15.66	4	155.75	29.91	32.15	60.52
12........	.250	.378	15.70	4	163.50	29.26	29.78	57.16
11........	.250	.367	15.74	4	171.75	28.45	26.07	51.33
10........	.250	.356	15.79	4	179.50	27.64	21.80	44.04

TABLE 66—Continued.

1	2	3	4	5	6	7	8	9
9........	.125	.257	16.03	4	111.50	20.06	16.92	46.41
8........	.125	.254	16.01	4	119.00	19.89	16.62	46.01
7........	.125	.252	16.07	4	126.75	19.76	16.16	44.88
6........	.125	.249	16.17	4	134.12	19.62	15.47	43.00
5........	.125	.247	16.23	4	141.00	19.45	14.55	40.65
4........	.125	.245	16.18	4	146.25	19.24	13.32	37.73
3........	.125	.242	16.22	4	152.25	19.07	12.02	34.26
2........	.125	.239	16.11	4	159.50	18.73	9.68	28.30
1........	.067	.161	16.49	4	153.25	12.80	....	....

Built by Swain Turbine and Manufacturing Co., Lowell, Mass.

TABLE 67.

Holyoke Test of a 45-inch Right Hand Samson Turbine. Test No. 979. Jan. 25 and 26, 1897. Conical Cylinder.

Number of experiment.	Gate opening (proportional part).	Proportional discharge (discharge at full gate with highest efficiency=1).	Mean head in feet.	Duration of test in minutes.	Revolutions per minute.	Discharge in second-feet.	Horse-power developed.	Percentage of efficiency.
1	2	3	4	5	6	7	8	9
8........	1.000	0.992	14.94	5	127.60	171.24	233.49	80.48
7........	1.000	1.000	14.88	5	133.40	172.12	236.84	81.54
6........	1.000	0.998	14.92	4	138.12	172.12	238.65	81.94
5........	1.000	0.999	15.00	4	144.00	172.69	240.97	82.03
4........	1.000	1.001	15.02	4	148.75	173.23	240.82	81.61
3........	1.000	1.002	15.03	3	153.33	173.38	239.89	81.18
2........	1.000	0.998	15.04	4	157.75	172.81	236.08	80.09
1........	1.000	0.986	15.11	3	169.33	171.11	218.85	74.64
18........	0.832	0.887	14.99	4	112.50	153.24	208.16	79.90
17........	0.832	0.892	15.02	4	119.75	154.34	215.05	81.80
16........	0.832	0.896	15.04	4	126.12	155.04	219.62	83.05
15........	0.832	0.897	15.03	4	132.25	155.27	223.11	84.30
14........	0.832	0.896	15.04	4	134.12	155.03	223.61	84.55
13........	0.832	0.893	15.06	4	143.00	154.74	221.79	83.92
12........	0.832	0.888	15.09	4	148.12	153.93	219.65	83.38
11........	0.832	0.881	15.16	4	151.25	153.12	214.01	81.29
10........	0.832	0.874	15.21	4	155.00	152.15	208.77	79.55
9........	0.832	0.847	15.32	4	160.50	148.02	196.52	76.42
27........	0.684	0.766	15.19	3	112.67	133.24	183.94	80.14
26........	0.684	0.769	15.12	3	121.33	133.52	189.83	82.91
25........	0.684	0.768	15.11	4	127.67	133.24	191.06	83.68
24........	0.684	0.762	15.14	4	131.50	132.34	187.85	82.67
23........	0.684	0.756	15.20	4	135.50	131.58	185.27	81.68
22........	0.684	0.745	15.28	4	133.00	130.06	182.49	80.97
21........	0.684	0.732	15.33	4	141.75	128.02	178.39	80.15
20........	0.684	0.728	15.39	4	147.00	127.52	176.99	79.52
19........	0.684	0.719	15.43	4	156.00	125.99	169.79	77.01
57........	0.568	0.641	15.85	5	125.80	113.89	162.59	79.42
58........	0.568	0.633	15.88	4	131.50	112.65	162.80	80.25
59........	0.568	0.630	15.85	4	135.75	112.04	160.68	79.78
60........	0.568	0.629	15.83	4	139.75	111.68	157.81	78.71
61........	0.568	0.622	15.84	4	143.25	110.45	152.99	77.11
62........	0.568	0.613	15.85	4	148.25	109.02	146.23	74.62
46........	0.424	0.500	16.50	4	112.50	90.70	123.97	73.05
45........	0.424	0.499	16.53	4	121.25	90.59	127.84	75.28
47........	0.424	0.499	16.49	4	124.00	90.37	127.79	75.61
44........	0.424	0.497	16.55	4	127.00	90.24	127.86	75.49
49........	0.424	0.497	16.47	4	126.87	90.04	127.73	75.95
43........	0.424	0.494	16.55	4	131.75	89.69	125.47	74.53
48........	0.424	0.487	16.50	4	135.50	88.24	121.67	73.69
42........	0.424	0.479	16.58	4	151.25	87.01	113.18	69.18

TABLE 68.

Holyoke Test of a 54-inch Right Hand Special Hercules Turbine. Built by the Holyoke Machine Co., Holyoke, Mass. Test No. 1051. Date Nov. 12, 1897.

Number of experiment.	Gate opening (proportional part).	Proportional discharge (discharge at full gate with highest efficiency=1).	Mean head in feet.	Duration of test in minutes.	Revolutions per minute.	Discharge in second-feet.	Horsepower developed.	Percentage of efficiency.
1	2	3	4	5	6	7	8	9
42........	1.000	1.004	13.98	5	80.40	230.06	305.38	83.72
41........	1.000	.997	14.07	4	84.12	228.98	306.94	84.00
40........	1.000	.989	14.13	4	87.00	227.80	305.62	83.71
39........	1.000	.981	14.22	4	90.50	226.71	305.62	83.58
38........	1.000	.974	14.26	4	94.00	225.30	303.39	83.26
37........	1.000	.964	14.29	4	98.00	223.16	299.65	82.83
36........	1.000	.956	14.34	4	101.50	221.65	293.21	81.31
35........	1.000	.944	14.38	4	104.87	219.38	285.03	79.66
34........	.800	.881	14.69	4	80.00	206.93	287.01	83.25
33........	.800	.875	14.73	5	83.20	205.72	287.19	83.56
32........	.800	.868	14.80	4	86.60	204.66	287.16	83.59
31........	.800	.859	14.87	4	90.25	202.98	286.38	83.65
30........	.800	.852	14.94	4	93.75	201.77	284.75	83.28
29........	.800	.844	15.01	4	97.12	200.41	281.78	82.59
28........	.800	.836	15.07	4	101.00	198.92	277.94	81.75
27........	.800	.829	15.09	4	104.87	197.26	270.78	80.20
26........	.800	.820	15.15	5	108.40	195.46	261.48	77.85
25........	.650	.749	15.38	5	77.00	179.88	246.95	78.70
24........	.650	.745	15.40	5	81.00	179.02	250.42	80.09
23........	.650	.739	15.45	5	84.60	177.87	251.78	80.78
22........	.650	.734	15.48	5	88.60	176.85	252.85	81.43
21........	.650	.728	15.49	5	92.80	175.59	252.22	81.76
20........	.650	.722	15.47	5	95.80	173.91	248.01	81.28
19........	.650	.714	15.50	4	99.25	172.09	242.10	80.02
18........	.650	.705	15.53	5	102.40	170.10	234.48	78.26
17........	.650	.696	15.57	4	105.75	168.14	226.34	76.23
16........	.527	.622	15.97	4	74.50	152.27	202.49	73.42
15........	.527	.618	16.01	4	78.87	151.47	205.79	74.82
14........	.527	.612	16.03	4	83.12	150.12	207.28	75.94
13........	.527	.606	16.09	4	87.25	148.80	207.50	76.41
12........	.527	.597	16.12	4	91.62	146.94	205.44	76.47
11........	.527	.591	16.15	4	95.37	145.60	200.89	75.32
10........	.527	.584	16.22	4	99.25	143.98	195.57	73.84
9........	.527	.578	16.23	4	103.00	142.55	188.96	72.01
8........	.410	.499	16.58	4	77.00	124.59	163.24	69.67
7........	.410	.494	16.63	4	81.62	123.44	163.61	70.27
6........	.410	.489	16.64	4	85.50	122.20	162.67	70.53
5........	.410	.483	16.68	4	90.50	120.82	159.88	69.95
4........	.410	.478	16.66	4	94.75	119.58	154.51	68.38
3........	.410	.472	16.68	5	99.10	118.20	148.14	66.25
2........	.410	.467	16.73	4	103.75	117.00	140.99	63.51
1........	.410	.460	16.79	5	109.30	115.53	129.97	59.07

TABLE 69.

Holyoke Test of a 36-inch Right Hand Victor Turbine. Test No. 1061, December 14, 1897. Conical Draft-tube. Cylinder Gate.

Number of experiment.	Gate opening (proportional part).	Proportional discharge (discharge at full gate with highest efficiency=1).	Mean head in feet.	Duration of test in minutes.	Revolutions per minute.	Discharge in second-feet.	Horsepower developed.	Percentage of efficiency.
1	2	3	4	5	6	7	8	9
12........	1.000	1.010	16.75	3	133.33	116.38	175.20	79.24
11........	1.000	1.009	16.76	4	139.25	116.38	176.99	80.01
10........	1.000	1.009	16.74	4	144.87	116.27	177.91	80.59
9........	1.000	1.002	16.79	4	150.75	115.63	177.73	80.71
8........	1.000	0.997	16.80	4	155.00	115.04	176.07	80.33
6........	1.000	0.992	16.73	4	156.50	114.30	172.97	79.75
7........	1.000	0.982	16.82	4	161.75	113.46	173.81	80.30
5........	1.000	0.975	16.69	4	162.75	112.21	169.89	79.98
4........	1.000	0.965	16.58	4	167.00	110.63	164.07	78.87
3........	1.000	0.953	16.59	4	172.25	109.31	158.65	77.14
2........	1.000	0.941	16.65	4	177.25	108.09	152.37	74.65
1........	1.000	0.923	16.70	4	184.00	106.21	141.23	70.20
53........	1.000	0.748	17.33	4	240.50	87.70	....	....
52........	0.900	0.967	16.92	4	133.75	112.07	172.47	80.19
51........	0.900	0.965	16.99	4	139.50	112.07	174.74	80.92
50........	0.900	0.959	17.01	4	144.00	111.37	174.19	81.07
49........	0.900	0.953	17.04	4	148.50	110.76	173.25	80.93
48........	0.900	0.947	17.03	4	152.00	110.03	171.73	80.81
47........	0.900	0.939	17.04	4	157.00	108.19	170.63	80.86
46........	0.900	0.925	17.05	4	164.00	107.62	168.17	80.81
45........	0.900	0.914	17.06	2	170.00	106.32	161.80	78.65
44........	0.801	0.900	17.10	3	132.33	104.79	160.07	78.76
43........	0.801	0.900	17.07	4	137.75	104.67	162.40	80.14
42........	0.801	0.899	17.02	4	142.25	104.32	162.46	80.64
41........	0.801	0.892	17.02	4	147.00	103.61	161.57	80.78
40........	0.801	0.884	17.02	4	151.25	102.68	159.74	80.59
39........	0.801	0.870	17.02	4	155.25	101.04	157.29	80.64
38........	0.801	0.863	16.97	4	159.25	100.09	154.50	80.20
37........	0.801	0.856	16.92	4	163.50	99.18	150.59	79.12
36........	0.801	0.845	16.89	4	168.25	97.79	144.64	77.21
35........	0.701	0.814	16.89	3	125.67	94.25	134.27	74.37
34........	0.701	0.814	16.90	4	133.25	94.25	139.09	76.99
33........	0.701	0.812	16.89	4	138.75	94.03	140.58	78.04
32........	0.701	0.807	16.92	4	144.25	93.47	140.83	78.51
31........	0.701	0.794	17.00	4	148.75	92.20	139.75	78.61
30........	0.701	0.787	17.07	5	153.60	91.54	137.70	77.70
29........	0.701	0.776	17.15	4	158.75	90.52	134.52	76.40
28........	0.701	0.768	17.15	4	163.25	89.52	130.31	74.84
27........	0.601	0.717	17.24	4	129.00	83.87	117.23	71.49
26........	0.601	0.714	17.29	4	137.00	83.65	120.29	73.33
25........	0.601	0.705	17.36	4	143.25	82.77	120.51	73.94
24........	0.601	0.696	17.44	4	148.00	81.91	119.05	73.48
23........	0.601	0.685	17.47	4	153.75	80.62	116.12	72.69
22........	0.601	0.676	17.53	4	159.00	79.77	112.28	70.79
21........	0.601	0.666	17.56	4	166.00	78.57	107.02	68.39
20........	0.502	0.622	17.60	3	123.33	78.47	96.93	66.09
19........	0.502	0.617	17.55	3	131.67	72.85	98.64	68.02
18........	0.502	0.609	17.55	3	138.00	71.81	98.29	68.77
17........	0.502	0.598	17.56	3	145.00	70.57	96.16	68.41
16........	0.502	0.592	17.55	4	150.25	69.87	93.18	67.00
15........	0.502	0.588	17.56	4	155.37	69.34	90.63	65.63
14........	0.502	0.580	17.56	3	163.50	68.44	85.34	62.61
13........	0.502	0.573	17.55	4	170.25	67.61	78.40	58.26

For experiment 53, the jacket was removed from the dynamometer.

TABLE 70.

Holyoke Test of a 57-inch Left Hand Jolly-McCormick Turbine. Conical Draft-tube. Test No. 1156. Oct. 31 and Nov. 1, 1898.

Number of experiment.	Gate opening (proportional part).	Proportional discharge (discharge at full gate with highest efficiency=1).	Mean head in feet.	Duration of test in minutes.	Revolutions per minute.	Discharge in second-feet.	Horse-power developed.	Percentage of efficiency.
1	2	3	4	5	6	7	8	9
40........	1.000	1.011	13.77	4	79.87	244.50	305.80	80.10
39........	1.000	1.007	13.86	4	83.25	244.34	309.12	80.49
38........	1.000	1.004	13.92	4	86.75	244.16	311.49	80.82
37........	1.000	1.001	13.93	4	90.25	243.38	311.78	81.10
36........	1.000	0.994	14.03	4	94.62	242.58	312.08	80.86
35........	1.000	0.984	14.10	4	99.87	240.66	305.63	79.43
34........	0.770	0.890	14.69	4	79.37	222.19	299.03	80.81
33........	0.770	0.886	14.72	4	82.87	221.41	302.63	81.89
32........	0.770	0.883	14.73	4	85.75	220.81	303.24	82.22
31........	0.770	0.881	14.75	4	88.75	220.38	303.58	82.36
30........	0.770	0.876	14.76	4	92.00	219.17	302.19	82.38
29........	0.770	0.868	14.82	4	95.50	217.65	298.75	81.68
28........	0.770	0.857	14.86	5	98.90	215.23	292.57	80.67
27........	0.770	0.847	14.94	5	101.50	213.41	283.00	78.28
26........	0.615	0.762	15.36	4	77.75	194.56	261.73	77.23
25........	0.615	0.761	15.36	4	81.87	194.25	266.69	78.82
24........	0.615	0.757	15.40	5	86.20	193.52	269.07	79.62
23........	0.615	0.753	15.39	4	89.62	192.50	267.55	79.64
22........	0.615	0.745	15.45	4	92.37	190.92	262.57	78.50
21........	0.615	0.739	15.47	4	95.62	189.30	257.51	77.54
20........	0.615	0.729	15.52	4	98.75	187.15	251.16	76.26
19........	0.615	0.719	15.63	4	102.25	185.14	243.37	74.17
18........	0.483	0.632	15.90	4	78.12	164.25	215.69	72.83
17........	0.483	0.630	15.87	4	82.37	163.42	218.46	74.28
16........	0.483	0.626	15.85	4	86.37	162.32	217.32	74.49
15........	0.483	0.621	15.81	4	89.50	160.80	213.03	73.90
14........	0.483	0.615	15.81	4	93.00	159.31	208.71	73.07
13........	0.483	0.609	15.74	4	96.00	157.40	202.38	72.04
12........	0.483	0.603	15.75	4	99.25	156.03	195.74	70.24
11........	0.483	0.598	15.72	4	102.37	154.55	187.97	68.23
10........	0.483	0.593	15.76	4	105.50	153.45	179.36	65.41
9........	0.360	0.500	16.27	4	74.75	131.52	161.14	66.41
8........	0.360	0.498	16.37	4	79.62	131.26	163.52	67.11
7........	0.360	0.495	16.42	4	83.75	130.63	163.46	67.20
6........	0.360	0.492	16.41	4	87.12	129.75	161.15	66.74
5........	0.360	0.488	16.43	4	90.37	128.87	157.94	65.78
4........	0.360	0.485	16.42	6	93.75	127.96	153.65	64.49
3........	0.360	0.481	16.45	4	96.62	127.21	149.15	62.86
2........	0.360	0.479	16.54	4	100.25	126.83	143.17	60.19
1........	0.360	0.474	16.59	4	104.37	125.70	134.86	57.03

TABLE 71.

Holyoke Test of a 45-inch Right Hand Victor Turbine. Test No. 1177, March 13 and 14, 1899. Conical Draft-tube.

Number of experi-ment.	Gate opening (propor-tional part).	Proportional discharge (discharge at full gate with highest efficiency=1).	Mean head in feet.	Dura-tion of test in min-utes.	Revolu-tions per minute.	Dis-charge in sec-ond-feet.	Horse-power devel-oped.	Percent-age of effi-ciency.
1	2	3	4	5	6	7	8	9
48........	1.000	1.012	15.22	4	102.37	180.69	252.39	80.92
47........	1.000	1.000	15.21	4	107.50	180.24	254.82	81.96
46........	1.000	1.004	15.20	4	111.50	179.26	253.70	82.10
45........	1.000	0.997	15.26	4	116.67	178.26	253.57	82.19
44........	1.000	0.986	15.31	5	121.60	176.54	251.08	81.91
43........	1.000	0.972	15.35	4	126.25	174.41	246.10	81.05
42........	1.000	0.954	15.36	5	128.40	171.19	235.46	78.96
41........	1.000	0.934	15.44	4	131.50	167.98	223.29	75.91
40........	0.900	0.959	15.32	4	98.37	171.76	239.19	80.15
39........	0.900	0.955	15.32	4	103.37	171.19	241.52	81.20
38........	0.900	0.949	15.36	4	107.25	170.19	240.39	81.08
37........	0.900	0.942	15.39	5	111.30	169.23	239.64	81.13
36........	0.900	0.929	15.50	5	115.80	167.41	238.31	80.98
35........	0.900	0.917	15.54	4	119.00	165.39	234.39	80.41
34........	0.900	0.907	15.58	4	122.50	163.86	230.47	79.60
33........	0.900	0.896	15.73	4	127.50	162.75	225.15	77.55
32........	0.900	0.890	15.73	4	133.50	161.52	217.62	75.52
30........	0.800	0.888	15.90	4	97.25	162.07	228.54	78.20
29........	0.800	0.886	16.04	5	105.70	162.46	238.35	80.65
28........	0.800	0.878	16.20	4	111.00	161.79	239.74	80.65
27........	0.800	0.867	16.34	4	118.00	160.41	241.24	81.15
31........	0.800	0.873	16.80	4	113.37	158.91	231.00	81.12
26........	0.800	0.861	16.24	4	131.35	157.14	232.71	80.32
25........	0.800	0.838	16.28	4	125.25	154.81	225.44	78.87
24........	0.800	0.826	16.20	4	131.87	152.24	214.96	76.85
23........	0.700	0.802	16.16	4	99.00	147.56	205.08	75.83
22........	0.700	0.799	16.15	4	104.12	147.04	207.20	76.94
21........	0.700	0.794	16.17	4	109.62	146.23	208.47	77.74
20........	0.700	0.781	16.12	4	114.50	143.56	206.09	78.52
19........	0.700	0.768	16.18	4	118.00	141.49	200.36	77.17
18........	0.700	0.758	16.19	4	122.75	139.54	184.26	75.82
17........	0.700	0.747	16.23	3	130.00	137.72	185.42	73.14
16........	0.600	0.701	16.39	4	96.00	129.89	169.53	70.21
15........	0.600	0.702	16.39	4	101.37	130.02	172.13	71.22
14........	0.600	0.696	16.38	4	106.75	128.88	173.29	72.38
13........	0.600	0.683	16.44	4	111.62	126.85	172.85	73.08
12........	0.600	0.676	16.44	4	115.50	125.45	170.23	72.78
11........	0.600	0.669	16.42	4	118.87	124.06	165.94	71.82
10........	0.600	0.662	16.43	4	124.50	122.81	160.66	70.21
9........	0.600	0.656	16.44	4	131.25	121.81	151.55	66.73
8........	0.502	0.607	16.60	4	95.25	113.21	139.09	65.26
7........	0.502	0.603	16.60	4	100.62	112.49	140.10	66.15
6........	0.502	0.594	16.64	4	104.62	110.91	139.27	66.54
5........	0.502	0.586	16.62	4	109.75	109.32	138.65	67.29
4........	0.502	0.580	16.62	5	114.80	108.25	136.45	66.87
3........	0.502	0.576	16.65	4	120.00	107.66	132.85	65.35
2........	0.502	0.574	16.68	4	125.50	107.30	127.86	62.99
1........	0.502	0.570	16.68	4	131.50	106.59	120.58	59.80

TABLE 72.

Holyoke Tests of a 38-inch Right Hand High Head Allis-Chalmers (Runner No. 20) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of: opening of the speed-gate. In inches.	Proportional part of: the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
91........	2.65	1.035	17.96	3	106.00	25.63	38.19	73.15
90........	2.65	1.027	17.96	3	116.00	25.44	39.65	76.51
89........	2.65	1.017	17.94	4	127.00	25.17	41.06	80.17
88........	2.65	1.008	17.94	4	135.50	24.95	41.31	81.36
87........	2.65	1.000	17.97	4	140.75	24.77	40.96	81.13
92........	2.65	1.002	17.97	4	141.50	24.83	41.17	81.36
86........	2.65	0.990	17.98	4	147.25	24.53	40.81	81.58
85........	2.65	0.972	18.02	4	155.25	24.13	39.44	79.97
84........	2.65	0.950	18.04	4	162.00	23.59	37.41	77.51
83........	2.65	0.932	17.99	4	171.00	23.11	35.54	75.37
82........	2.42	0.979	17.65	4	112.25	24.04	37.85	78.65
81........	2.42	0.963	17.74	4	128.50	23.71	39.77	83.36
79........	2.42	0.952	17.73	4	136.50	23.44	39.72	84.27
80........	2.42	0.948	17.72	4	142.00	23.32	39.68	84.66
78........	2.42	0.939	17.73	4	146.75	23.11	39.31	84.59
77........	2.42	0.927	17.76	4	152.25	22.84	38.68	84.07
76........	2.42	0.907	17.79	3	157.67	22.36	36.41	80.71
75........	2.42	0.889	17.82	4	166.50	21.94	34.61	78.04
11........	2.125	0.912	17.73	4	102.25	22.44	34.95	77.45
9........	2.125	0.899	17.76	4	115.50	22.14	36.81	82.54
8........	2.125	0.890	17.76	4	123.00	21.93	37.50	84.88
7........	2.125	0.882	17.75	4	129.75	21.72	37.75	86.34
6........	2.125	0.872	17.78	4	136.75	21.49	37.90	87.45
10........	2.125	0.866	17.79	4	142.50	21.34	37.84	87.89
5........	2.125	0.854	17.77	4	146.25	21.04	37.15	87.61
3........	2.125	0.837	17.81	4	154.25	20.64	35.62	85.44
4........	2.125	0.821	17.83	4	163.00	20.26	33.88	82.69
2........	2.125	0.797	17.86	4	170.50	19.69	31.50	78.98
12........	2.125	0.741	17.86	4	180.75	18.30	25.05	67.56
1........	2.125	0.460	18.21	4	222.50	11.48	....	....
22........	1.875	0.855	17.67	4	97.75	21.01	32.51	77.20
20........	1.875	0.848	17.69	4	105.75	20.84	33.70	80.60
19........	1.875	0.836	17.70	4	115.50	20.55	34.69	84.05
18........	1.875	0.822	17.72	4	128.50	20.23	35.61	87.59
21........	1.875	0.816	17.70	4	134.00	20.06	35.59	88.37
17........	1.875	0.804	17.74	4	138.50	19.80	35.18	88.32
15........	1.875	0.787	17.75	4	148.25	19.37	34.24	87.80
16........	1.875	0.770	17.75	3	158.00	18.97	32.84	85.99
23........	1.875	0.769	17.79	4	157.75	18.97	32.79	85.66
14........	1.875	0.748	17.79	4	164.25	18.44	30.35	81.56
13........	1.875	0.698	17.85	4	178.25	17.24	24.70	70.76
73........	1.84	0.836	17.90	3	107.67	20.68	33.82	80.55
72........	1.84	0.825	17.93	4	118.50	20.42	35.03	84.35
71........	1.84	0.813	17.94	4	129.25	20.13	35.82	87.45
70........	1.84	0.807	17.94	4	133.25	19.99	35.70	87.76
69........	1.84	0.802	17.89	4	136.25	19.84	35.24	87.54
74........	1.84	0.802	17.91	4	136.50	19.84	35.31	87.60
68........	1.84	0.792	17.93	4	141.25	19.61	35.23	88.34
67........	1.84	0.789	17.88	4	143.00	19.50	34.35	86.85
66........	1.84	0.775	17.90	4	152.25	19.61	33.75	86.77
65........	1.84	0.755	17.94	4	160.25	18.68	31.83	83.74
64........	1.84	0.734	17.95	4	167.00	18.17	29.31	79.24
63........	1.84	0.711	18.00	4	174.50	17.62	26.60	73.94

TABLE 72—Continued.

1	2	3	4	5	6	7	8	9
35........	1.625	0.778	17.87	5	102.60	19.22	31.28	80.29
34........	1.625	0.769	17.80	4	108.25	18.97	31.50	82.25
33........	1.625	0.757	17.81	4	117.75	18.68	32.09	85.04
32........	1.625	0.745	17.82	4	128.50	18.38	32.64	87.87
37........	1.625	0.743	18.06	4	130.25	18.46	33.09	87.50
31........	1.625	0.739	17.85	4	132.00	18.26	32.31	87.41
39........	1.625	0.733	17.94	4	135.50	18.15	32.54	88.12
29........	1.625	0.727	17.90	4	139.00	17.98	32.10	87.94
38........	1.625	0.728	17.97	4	140.00	18.04	32.33	87.93
30........	1.625	0.719	17.86	4	145.00	17.76	31.48	87.50
28........	1.625	0.707	17.93	5	150.20	17.49	31.22	87.77
36........	1.625	0.709	18.10	4	151.00	17.62	31.38	86.77
27........	1.625	0.687	17.96	4	158.50	17.02	29.28	84.46
26........	1.625	0.666	17.97	4	166.75	16.51	26.96	80.11
25........	1.625	0.643	18.01	4	173.75	15.94	24.08	73.94
24........	1.625	0.616	18.06	4	181.50	15.30	20.96	66.87
51........	1.375	0.664	17.95	4	99.00	16.45	26.52	79.19
50........	1.375	0.657	17.91	3	106.00	16.26	26.93	81.53
49........	1.375	0.649	17.90	4	113.00	16.05	27.14	83.29
48........	1.375	0.639	17.91	4	120.50	15.81	27.27	84.92
47........	1.375	0.628	17.92	4	127.25	15.54	27.04	85.60
46........	1.375	0.619	17.97	4	133.50	15.33	26.51	84.86
45........	1.375	0.606	17.98	4	139.50	15.01	25.77	84.20
44........	1.375	0.598	18.07	4	144.25	14.86	25.32	83.13
43........	1.375	0.583	18.07	4	151.00	14.49	24.41	82.20
42........	1.375	0.563	18.00	4	161.50	13.97	22.38	78.46
41........	1.375	0.542	18.06	4	171.75	13.47	19.83	71.88
40........	1.375	0.515	18.10	4	182.75	12.81	16.88	64.19
61........	1.115	0.562	18.09	3	97.33	13.97	22.48	78.42
60........	1.115	0.555	18.07	3	105.00	13.79	22.79	80.65
59........	1.115	0.543	18.07	4	115.50	13.49	22.94	82.97
58........	1.115	0.531	18.09	3	122.33	13.21	22.60	83.38
57........	1.115	0.521	18.10	4	128.50	12.96	21.96	82.54
62........	1.115	0.522	18.12	3	129.00	12.98	22.05	82.64
56........	1.115	0.510	18.12	4	135.75	12.68	21.32	81.81
55........	1.115	0.497	18.14	4	142.25	12.38	20.37	79.96
54........	1.115	0.486	18.14	3	150.00	12.09	19.40	77.99
53........	1.115	0.472	18.17	4	158.25	11.77	18.27	75.33
52........	1.115	0.453	18.14	4	170.75	11.27	15.77	68.03

TABLE 73.

Holyoke Tests of a 55 3/4-inch Right Hand Special High Head S. Morgan-Smith Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
....	1.000	0.794	17.75+	4	Still	46.10	....	....
11........	1.000	0.989	17.50	4	79.00	57.00	84.44	74.64
7........	1.000	0.997	17.42	4	87.50	57.33	90.99	80.34
8........	1.000	1.000	17.50	4	91.87	57.66	94.48	82.56
9........	1.000	1.001	17.56	3	95.00	57.79	96.05	83.46
10........	1.000	1.001	17.54	4	97.00	57.79	96.95	84.34
6........	1.000	1.002	17.39	4	98.75	57.59	96.99	85.39
5........	1.000	0.978	17.42	4	109.00	56.27	94.46	84.97
4........	1.000	0.962	17.46	4	119.25	55.42	89.56	81.61
3........	1.000	0.913	17.52	3	129.00	52.64	74.53	71.25
2........	1.000	0.789	17.68	4	137.50	45.73	39.72	43.32
1........	1.000	0.590	17.90	4	144.50	34.39	....	....
21........	0.917	0.951	17.49	4	82.00	54.83	86.69	79.71
18........	0.917	0.953	17.44	4	85.50	54.83	88.91	81.99
19........	0.917	0.954	17.45	5	88.80	54.90	90.81	83.38
20........	0.917	0.953	17.49	4	90.50	54.90	92.02	84.50
17........	0.917	0.952	17.44	4	95.50	54.77	93.79	86.58
16........	0.917	0.940	17.46	3	100.00	54.11	92.44	86.27
15........	0.917	0.928	17.45	4	105.25	53.41	91.21	86.29
14........	0.917	0.891	17.49	4	118.50	51.36	82.15	80.64
13........	0.917	0.840	17.55	4	127.50	48.52	66.29	68.65
12........	0.917	0.738	17.70	4	133.50	42.81	38.56	44.88
33........	0.833	0.906	17.54	3	77.00	52.31	82.30	79.09
32........	0.833	0.905	17.55	4	82.25	52.25	85.53	82.25
30........	0.833	0.904	17.58	4	87.50	52.25	88.47	84.92
31........	0.833	0.902	17.58	4	90.50	52.12	89.93	86.54
29........	0.833	0.900	17.58	4	91.50	52.00	89.87	86.68
28........	0.833	0.894	17.54	4	92.75	51.61	88.41	86.12
27........	0.833	0.887	17.56	4	95.50	51.23	88.28	86.53
26........	0.833	0.875	17.56	4	101.00	50.53	87.53	86.98
25........	0.833	0.860	17.56	4	109.25	49.65	85.21	86.18
24........	0.833	0.835	17.66	4	114.00	48.33	79.03	81.05
23........	0.833	0.780	17.71	4	122.75	45.24	63.82	70.24
22........	0.833	0.687	17.84	4	130.25	39.96	37.62	46.54
44........	0.758	0.840	17.55	4	74.50	48.52	77.47	80.22
43........	0.758	0.837	17.55	3	79.33	48.33	80.20	83.38
42........	0.758	0.829	17.58	3	85.00	47.90	82.01	85.87
41........	0.758	0.821	17.60	4	87.75	47.46	82.13	86.70
40........	0.758	0.816	17.60	4	89.75	47.15	81.41	86.50
39........	0.758	0.812	17.65	4	91.50	47.03	81.94	87.04
38........	0.758	0.796	17.67	3	99.00	46.10	80.07	86.68
37........	0.758	0.772	17.72	3	105.00	44.81	75.83	84.20
36........	0.758	0.747	17.78	3	110.00	43.42	69.91	79.84
35........	0.758	0.690	17.94	4	119.50	40.25	55.23	67.44
34........	0.768	0.619	18.02	4	125.00	36.20	36.11	48.81

TABLE 73—Continued.

1	2	3	4	5	6	7	8	9
57........	0.667	0.784	16.84	3	72.00	44.33	69.05	81.56
56........	0.667	0.777	16.99	3	78.00	44.14	72.10	84.77
55........	0.667	0.768	17.20	3	82.00	43.90	73.43	85.75
53........	0.667	0.757	17.56	4	87.25	43.72	75.10	86.26
54........	0.667	0.757	17.39	4	88.50	43.48	74.65	87.05
51........	0.667	0.751	17.75	4	91.50	43.60	76.65	87.32
52........	0.667	0.746	17.72	4	92.75	43.29	76.09	87.46
50........	0.667	0.736	17.77	4	95.50	42.75	74.48	86.45
49........	0.667	0.721	17.73	4	99.25	41.86	71.67	85.15
48........	0.667	0.696	17.78	4	104.25	40.43	66.24	81.26
47........	0.667	0.669	17.85	4	109.75	38.96	60.24	76.37
46........	0.667	0.643	17.86	4	114.50	37.46	52.92	69.75
45........	0.667	0.577	18.00	4	121.25	33.72	35.02	50.88
71........	0.575	0.686	17.57	4	68.50	39.61	61.34	77.72
70........	0.575	0.686	17.57	4	75.50	39.61	65.43	82.90
69........	0.575	0.678	17.34	4	81.00	38.91	65.51	85.62
68........	0.575	0.666	17.29	3	85.00	38.15	63.84	85.34
67........	0.575	0.656	17.12	4	87.00	37.40	61.82	85.14
66........	0.575	0.648	17.17	4	89.25	37.00	60.84	84.45
64........	0.575	0.652	16.32	4	85.50	36.31	56.81	84.52
63........	0.575	0.642	16.49	4	88.75	35.91	56.40	83.99
65........	0.575	0.638	17.27	4	91.25	36.54	59.57	83.24
62........	0.575	0.633	16.57	4	90.75	35.51	55.05	82.50
61........	0.575	0.624	16.68	4	93.00	35.12	53.73	80.87
59........	0.575	0.601	17.13	4	99.00	34.28	51.48	77.30
60........	0.575	0.575	16.91	4	103.75	32.56	44.95	71.99
58........	0.575	0.521	17.41	4	115.00	29.95	33.22	56.18
79........	0.479	0.585	17.81	4	66.75	34.05	52.06	75.70
78........	0.479	0.579	17.80	3	77.00	33.67	55.61	81.81
77........	0.479	0.567	17.83	3	83.00	33.00	55.14	82.64
76........	0.479	0.553	17.88	3	87.00	32.23	52.78	80.75
75........	0.479	0.539	17.89	4	90.50	31.41	50.19	78.76
74........	0.479	0.528	17.90	3	92.67	30.81	48.18	77.04
73........	0.479	0.513	17.88	4	97.40	29.89	45.02	74.27
80........	0.479	0.494	17.99	4	101.75	28.88	41.15	69.84
72........	0.479	0.453	17.96	4	108.25	26.48	31.27	57.98
90........	0.408	0.470	17.80	3	68.00	27.31	41.25	74.82
89........	0.408	0.463	17.84	4	70.75	26.94	40.87	75.00
88........	0.408	0.453	17.87	3	75.00	26.38	40.30	75.37
87........	0.408	0.442	17.88	4	79.25	25.76	38.92	74.50
86........	0.408	0.430	17.96	4	83.50	25.10	37.14	72.66
85........	0.408	0.417	18.05	4	86.50	24.44	34.98	69.92
84........	0.408	0.407	18.10	3	91.00	23.89	33.12	67.54
83........	0.408	0.403	18.10	4	92.75	23.60	32.15	66.37
82........	0.408	0.383	18.10	4	99.25	22.47	28.67	62.16
81........	0.408	0.343	18.12	4	112.75	20.11	19.54	47.29

TABLE 74.

Holyoke Tests of a 55.25-inch Right Hand Wellman-Seaver-Morgan Co. (Runner No. 18) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
11........	1.000	1.046	17.70	3	66.67	64.25	92.47	71.85
10........	1.000	1.027	17.75	3	78.33	63.16	99.59	78.50
9........	1.000	1.018	17.77	4	85.00	62.62	103.16	81.92
7........	1.000	1.010	17.77	4	91.50	62.14	105.76	84.63
6........	1.000	0.994	17.79	3	98.00	61.20	104.78	85.04
5........	1.000	0.968	17.83	4	102.50	59.65	100.70	83.67
8........	1.000	0.937	17.87	4	108.25	57.79	93.84	80.30
4........	1.000	0.937	17.87	3	108.00	57.79	93.62	80.11
3........	1.000	0.913	17.87	4	118.00	56.34	88.65	77.81
12........	1.000	0.913	17.82	4	117.25	56.27	88.09	77.63
2........	1.000	0.881	17.85	4	125.75	54.31	79.94	72.87
13........	1.000	0.885	17.83	4	125.75	54.51	79.94	72.68
1........	1.000	0.526	18.32	4	156.00	32.84	....	....
22........	0.909	0.993	17.65	4	66.50	60.86	90.32	74.30
21........	0.909	0.973	17.70	3	78.00	59.72	96.92	81.02
20........	0.909	0.958	17.71	3	88.00	58.85	101.71	86.24
19........	0.909	0.942	17.73	3	95.00	57.86	101.57	87.49
18........	0.909	0.918	17.76	3	100.00	56.47	98.25	86.56
17........	0.909	0.889	17.82	3	107.00	54.77	92.76	83.98
16........	0.909	0.864	17.85	4	115.75	53.28	86.96	80.80
15........	0.909	0.829	17.89	3	123.67	51.17	78.62	75.89
14........	0.909	0.496	18.37	4	157.00	31.03	....	....
31........	0.777	0.897	17.83	3	65.00	55.29	84.52	75.76
30........	0.777	0.882	17.83	4	72.75	54.38	88.29	80.47
29........	0.777	0.869	17.87	4	82.50	53.60	92.97	85.77
28........	0.777	0.847	17.90	4	89.50	52.31	93.10	87.86
27........	0.777	0.826	17.92	4	96.25	51.04	91.78	88.67
26........	0.777	0.808	17.90	4	103.25	49.90	89.51	88.55
25........	0.777	0.779	17.94	4	111.00	48.15	83.39	85.31
32........	0.777	0.766	17.99	4	114.00	47.40	79.06	81.93
24........	0.777	0.730	18.02	4	121.25	45.24	70.07	75.95
23........	0.777	0.453	18.43	3	157.67	28.35	....	....
43........	0.646	0.800	17.96	4	63.00	49.46	76.46	76.06
42........	0.646	0.784	18.00	4	72.00	48.52	81.14	82.10
40........	0.646	0.769	18.03	3	81.00	47.65	84.26	86.67
39........	0.646	0.747	18.03	3	87.00	46.28	82.96	87.85
38........	0.646	0.730	18.05	4	95.25	45.24	82.57	89.35
41........	0.646	0.718	18.10	4	99.75	44.57	81.86	89.67
37........	0.646	0.708	18.06	4	102.50	43.90	79.97	89.13
36........	0.646	0.678	18.14	4	111.25	42.15	73.94	85.45
35........	0.646	0.646	18.20	3	118.00	40.20	64.79	78.25
34........	0.646	0.607	18.21	4	125.25	37.80	54.29	69.69
33........	0.646	0.408	18.50	4	157.50	25.61	....	....
51........	0.518	0.697	18.11	4	67.75	43.29	70.48	79.44
50........	0.518	0.679	18.13	3	77.00	42.21	73.43	84.78
49........	0.518	0.661	18.17	4	85.25	41.14	73.90	87.36
48........	0.518	0.640	18.22	3	93.00	39.84	72.56	88.33

TABLE 74—Continued.

1	2	3	4	5	6	7	8	9
47........	0.518	0.626	18.21	4	97.50	38.96	70.43	87.73
46........	0.518	0.603	18.26	4	105.25	37.63	66.91	86.05
45........	0.518	0.572	18.29	4	114.25	35.68	59.42	80.47
44........	0.518	0.517	18.34	4	127.50	32.34	44.21	65.87
59........	0.390	0.543	18.33	4	66.25	33.94	55.52	78.85
58........	0.390	0.528	18.37	4	76.50	33.00	57.47	83.78
57........	0.390	0.507	18.39	4	84.50	31.74	56.16	85.02
56........	0.390	0.493	18.38	4	89.75	30.87	54.46	84.82
54........	0.390	0.472	18.41	4	98.50	29.57	51.23	83.16
53........	0.390	0.442	18.50	4	111.50	27.77	45.11	77.58
55........	0.390	0.419	18.48	4	120.25	26.27	38.22	69.57
52........	0.390	0.389	18.57	4	130.50	24.49	30.17	58.62

TABLE 75.

Holyoke Tests of a 34.75-inch Right Hand I. P. Morris (Runner "N") Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of opening of the speed-gate. In inches.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
75........	4.20	1.100	17.46	3	132.67	45.72	68.27	75.41
74........	4.20	1.092	17.50	3	145.67	45.44	69.67	77.25
73........	4.20	1.085	17.58	4	155.25	45.22	69.55	77.14
72........	4.20	1.080	17.62	4	162.00	45.08	68.65	76.21
71........	4.20	1.067	17.63	4	169.75	44.53	66.80	75.02
70........	4.20	1.050	17.62	4	174.50	43.81	63.39	72.40
69........	4.20	0.991	17.61	4	195.50	41.36	53.26	64.48
68........	4.20	0.941	17.63	4	228.00	39.29	41.41	52.71
37........	3.40	1.043	17.58	4	99.25	43.50	61.29	70.66
36........	3.40	1.028	17.66	4	124.00	42.96	69.06	80.27
35........	3.40	1.022	17.67	4	131.25	42.74	68.33	79.78
34........	3.40	1.006	17.68	4	142.25	42.07	69.76	82.69
33........	3.40	1.001	17.68	4	155.00	41.85	71.32	84.99
32........	3.40	0.986	17.70	4	164.50	41.27	69.71	84.15
31........	3.40	0.972	17.72	4	172.00	40.70	67.68	82.75
30........	3.40	0.958	17.71	4	178.75	40.08	64.93	80.65
29........	3.40	0.946	17.62	4	185.00	39.47	61.60	78.10
28........	3.40	0.915	17.62	4	202.25	38.21	55.10	72.16
27........	3.40	0.857	17.68	4	232.75	35.85	42.27	58.81
26........	3.00	0.965	17.49	4	99.00	40.13	58.14	73.03
25........	3.00	0.956	17.47	4	115.25	39.73	62.80	79.77
24........	3.00	0.942	17.48	3	122.00	39.16	62.78	80.87
23........	3.00	0.937	17.49	3	136.00	38.95	65.87	85.25
22........	3.00	0.928	17.51	4	147.25	38.60	66.86	87.22
21........	3.00	0.921	17.53	4	153.50	38.34	66.44	87.17
20........	3.00	0.915	17.55	4	156.50	38.12	66.32	87.41
19........	3.00	0.902	17.52	4	164.75	37.56	64.83	86.87
18........	3.00	0.889	17.54	4	173.75	37.01	63.11	85.72
17........	3.00	0.868	17.54	4	193.00	36.15	58.42	81.24
16........	3.00	0.808	17.60	4	217.25	33.71	46.03	68.40
15........	3.00	0.765	17.67	4	236.25	31.98	35.76	55.79
13........	2.50	0.843	17.56	3	103.67	35.14	53.35	76.23
12........	2.50	0.836	17.56	4	125.75	34.84	58.62	84.47
11........	2.50	0.826	17.57	4	137.00	34.42	59.72	87.07
9........	2.50	0.812	17.61	4	142.75	33.88	58.77	86.85
10........	2.50	0.809	17.56	4	148.25	33.71	58.79	87.56
14........	2.50	0.805	17.61	6	156.00	33.59	59.50	88.69
8........	2.50	0.800	17.68	4	155.25	33.46	59.21	88.25
7........	2.50	0.790	17.83	5	161.00	33.17	58.48	87.19
6........	2.50	0.779	17.84	4	170.00	32.72	56.61	85.50
4........	2.50	0.760	17.68	5	174.20	31.78	52.73	82.75
3........	2.50	0.728	17.71	4	192.00	30.48	46.49	75.95
2........	2.50	0.688	17.75	4	207.50	28.81	37.69	64.98
1........	2.50	0.641	17.83	6	228.67	26.92	27.69	50.86
49........	2.00	0.696	17.90	4	106.50	29.28	45.13	75.93
46........	2.00	0.696	17.78	4	122.50	29.20	48.21	81.87
48........	2.00	0.691	17.79	4	132.50	29.00	49.73	85.00
45........	2.00	0.688	17.79	4	138.25	28.85	50.22	86.27
44........	2.00	0.674	17.83	4	147.75	28.29	49.20	86.00
50........	2.00	0.664	17.97	4	155.75	28.01	49.03	85.89
47........	2.00	0.652	17.82	4	160.25	27.38	46.57	84.15
43........	2.00	0.639	17.89	4	165.50	26.88	45.09	82.67
40........	2.00	0.619	18.06	4	175.00	26.14	42.38	79.15
39........	2.00	0.585	18.08	4	196.00	24.73	35.60	70.20
38........	2.00	0.524	18.10	4	227.75	22.18	20.60	45.42

TABLE 75—Continued.

1	2	3	4	5	6	7	8	9
58........	1.50	0.570	18.19	4	107.75	24.17	39.14	78.49
59........	1.50	0.567	18.12	4	114.75	23.99	39.60	80.32
57........	1.50	0.562	18.28	4	123.50	23.91	41.12	82.96
56........	1.50	0.557	18.26	4	136.00	23.65	41.17	84.05
55........	1.50	0.547	18.27	3	147.67	23.25	40.23	83.51
54........	1.50	0.530	18.26	4	159.75	22.54	38.69	82.87
53........	1.50	0.513	18.26	4	172.50	21.78	36.55	81.04
52........	1.50	0.479	18.28	4	199.00	20.37	30.12	71.32
51........	1.50	0.443	18.33	5	223.60	18.86	20.31	51.79
67........	0.75	0.316	18.31	3	87.33	13.45	18.50	66.25
66........	0.75	0.312	18.33	3	112.00	13.30	20.34	73.57
65........	0.75	0.304	18.38	4	124.75	12.96	19.64	72.68
62........	0.75	0.298	18.40	5	134.40	12.71	18.71	70.56
63........	0.75	0.285	18.43	4	151.50	12.16	17.43	68.55
64........	0.75	0.279	18.41	4	161.50	11.89	17.11	68.92
61........	0.75	0.269	18.48	4	181.50	11.50	16.48	68.38
60........	0.75	0.216	18.57	4	239.00	9.27	....	....
82........	4.20	0.781	18.14	3	281.33	33.09	....	....
81........	3.40	0.680	18.23	3	282.00	28.88	....	....
80........	3.00	0.612	18.22	4	279.75	25.99	....	....
79........	2.50	0.542	18.06	4	276.25	22.91	....	....
78........	2.00	0.449	18.21	4	273.25	19.04	....	....
77........	1.50	0.366	18.24	4	268.50	15.55	....	....
76........	0.75	0.212	18.46	4	238.75	9.07	....	....

TABLE 76.

Holyoke Tests of a 32-inch Right Hand Wellman-Seaver-Morgan Co. Turbine. Wheel supported by Ball-bearing step. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
69........	1.000	1.024	17.23	4	112.50	34.45	46.84	69.59
67........	1.000	1.009	17.50	3	152.67	34.22	55.80	82.16
68........	1.000	1.008	17.57	2	163.00	34.25	57.31	83.98
66........	1.000	1.005	17.46	3	164.00	34.05	56.90	84.40
65........	1.000	0.992	17.47	3	172.67	33.60	55.92	84.00
64........	1.000	0.979	17.48	3	181.33	33.17	54.53	82.93
63........	1.000	0.965	17.46	4	189.25	32.67	52.53	81.21
62........	1.000	0.946	17.47	4	213.00	32.06	49.27	77.57
61........	1.000	0.688	17.59	4	251.50	23.39	34.91	74.81
60........	1.000	0.631	17.86	4	298.00	21.62	....	....
59........	0.889	0.933	17.50	3	113.67	31.63	45.23	72.04
58........	0.889	0.934	17.52	3	135.67	31.70	50.84	80.72
57........	0.889	0.931	17.54	3	146.33	31.60	52.80	84.01
56........	0.889	0.925	17.53	3	153.67	31.40	53.32	85.41
55........	0.889	0.922	17.44	4	156.50	31.21	52.85	85.62
54........	0.889	0.916	17.45	3	160.33	31.00	52.66	85.84
53........	0.889	0.909	17.46	3	164.00	30.79	52.35	85.87
52........	0.889	0.903	17.47	4	168.00	30.59	52.07	85.92
51........	0.889	0.898	17.48	4	172.75	30.42	51.95	86.14
50........	0.889	0.893	17.48	4	176.75	30.25	51.52	85.91
49........	0.889	0.886	17.49	4	184.00	30.05	51.08	85.69
48........	0.889	0.864	17.53	4	206.50	29.31	47.77	81.98
47........	0.889	0.795	17.59	4	244.75	27.02	33.97	63.02
46........	0.889	0.572	17.91	4	298.75	19.61	....	....
45........	0.741	0.809	17.57	4	106.00	27.48	39.23	71.65
44........	0.741	0.801	17.57	4	139.50	27.20	45.18	83.35
43........	0.741	0.791	17.55	3	152.00	26.85	45.71	85.53
41........	0.741	0.784	17.57	4	159.75	26.63	45.82	86.35
40........	0.741	0.780	17.60	4	166.25	26.52	46.15	87.18
42........	0.741	0.777	17.56	4	171.50	26.40	46.02	87.53
39........	0.741	0.770	17.64	4	180.75	26.22	45.99	87.68
38........	0.741	0.758	17.67	4	191.50	25.82	44.30	85.61
37........	0.741	0.727	17.73	3	210.67	24.81	38.99	78.15
36........	0.741	0.693	17.79	3	231.33	23.68	32.11	67.20
35........	0.741	0.503	18.03	2	298.00	17.31	....	....
34........	0.593	0.669	17.84	3	101.00	22.89	32.71	70.63
33........	0.593	0.660	17.84	3	122.67	22.61	36.32	79.40
31........	0.593	0.644	17.86	3	145.00	22.07	37.57	84.04
30........	0.593	0.642	17.85	3	151.33	21.98	37.81	84.97
32........	0.593	0.637	17.88	4	158.00	21.83	38.01	85.87
29........	0.593	0.632	17.91	4	163.75	21.68	37.88	86.02
28........	0.593	0.622	17.91	3	173.00	21.33	36.82	84.98
27........	0.593	0.606	17.99	3	188.00	20.82	34.79	81.90
26........	0.593	0.537	18.16	4	239.00	18.54	22.11	57.92
25........	0.593	0.406	18.05	3	281.00	13.98	....	....
24........	0.444	0.505	17.91	3	94.00	17.32	23.92	67.99
22........	0.444	0.489	17.92	2	130.00	16.78	27.06	79.36
21........	0.444	0.486	17.94	3	136.00	16.70	27.68	81.48
23........	0.444	0.486	17.92	4	139.25	16.66	27.70	81.82
20........	0.444	0.481	17.94	3	145.33	16.50	27.57	82.12
19........	0.444	0.472	17.94	4	153.00	16.21	26.90	81.56
18........	0.444	0.465	17.95	5	161.40	15.98	26.13	80.34
17........	0.444	0.450	17.99	4	178.00	15.46	24.71	78.32
16........	0.444	0.402	18.04	4	231.50	13.85	16.07	56.70
15........	0.444	0.325	18.17	4	281.25	11.22	....	....

TABLE 76—Continued.

1	2	3	4	5	6	7	8	9
14........	0.296	0.325	18.19	3	89.67	11.22	15.35	66.32
13........	0.296	0.320	18.21	3	107.00	11.06	16.58	72.60
11........	0.296	0.317	18.23	3	114.67	10.97	16.98	74.85
12........	0.296	0.316	18.22	3	118.33	10.94	16.97	75.07
10........	0.296	0.313	18.24	3	122.67	10.82	17.03	76.07
9........	0.296	0.308	18.25	4	130.75	10.68	16.94	76.62
8........	0.296	0.304	18.26	4	137.75	10.54	16.57	75.91
7........	0.296	0.300	18.30	4	146.50	10.41	16.27	75.29
6........	0.296	0.295	18.31	4	154.75	10.23	15.75	74.15
5........	0.296	0.291	18.32	4	164.00	10.10	15.17	72.31
4........	0.296	0.283	18.33	5	181.20	9.82	14.25	69.81
3........	0.296	0.276	18.36	4	191.25	9.58	13.27	66.53
2........	0.296	0.266	18.33	4	208.00	9.23	11.55	60.18
1........	0.296	0.225	18.37	4	261.00	7.83	....	....

TABLE 77.

Holyoke Test of a 45-inch Right Hand Special McCormick's Turbine Wheel.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
10........	1.000	1.031	16.37	4	100.50	112.94	173.92	83.09
9........	1.000	1.025	16.41	4	104.87	112.45	175.02	83.78
8........	1.000	1.019	16.46	4	109.50	111.96	176.00	84.36
7........	1.000	1.013	16.50	5	114.00	111.35	176.22	84.72
6........	1.000	1.007	16.23	4	116.00	109.78	172.16	85.35
5........	1.000	1.000	16.23	4	120.37	109.05	171.24	85.46
4........	1.000	0.994	16.23	4	124.50	108.45	169.45	85.04
3........	1.000	0.987	16.28	4	129.50	107.85	167.48	84.26
2........	1.000	0.978	16.31	5	134.80	106.91	163.54	82.85
1........	1.000	0.966	16.37	4	141.50	105.84	156.85	79.97
18........	0.783	0.913	16.63	4	96.00	100.80	151.94	80.07
17........	0.783	0.906	16.64	4	101.50	100.09	154.39	81.89
16........	0.783	0.899	16.67	4	106.75	99.40	155.81	83.06
15........	0.783	0.894	16.68	4	111.50	98.83	155.87	83.52
14........	0.783	0.888	16.67	4	115.50	98.14	154.35	83.34
13........	0.783	0.882	16.67	4	120.00	97.46	152.97	83.17
12........	0.783	0.875	16.65	4	125.50	96.67	150.71	82.71
11........	0.783	0.866	16.68	4	132.00	95.74	146.32	80.94
25........	0.612	0.777	17.16	4	97.25	87.17	129.96	76.75
24........	0.612	0.771	17.17	4	103.00	86.51	131.30	78.08
23........	0.612	0.765	17.18	4	108.75	85.85	131.94	79.02
22........	0.612	0.758	17.17	4	113.75	85.08	131.00	79.21
21........	0.612	0.751	17.14	4	118.75	84.21	128.71	78.77
20........	0.612	0.746	16.97	4	122.75	83.22	124.73	78.02
19........	0.612	0.735	17.00	4	129.50	82.02	119.63	75.79
33........	0.503	0.676	17.29	4	95.00	76.15	107.65	72.22
32........	0.503	0.672	17.30	4	101.25	75.62	109.12	73.68
31........	0.503	0.665	17.31	4	107.00	74.88	109.39	74.55
30........	0.503	0.658	17.34	4	113.50	74.13	109.04	74.93
29........	0.503	0.651	17.36	4	118.50	73.40	107.28	74.37
28........	0.503	0.643	17.38	4	123.75	72.56	104.41	73.13
27........	0.503	0.633	17.39	5	130.00	71.41	100.07	71.19
26........	0.503	0.620	17.41	4	137.75	69.99	93.32	67.65
41........	0.373	0.548	17.56	4	93.00	62.15	81.33	65.83
40........	0.373	0.545	17.54	4	98.25	61.76	82.29	67.10
39........	0.373	0.540	17.59	4	104.50	61.35	83.02	67.95
38........	0.373	0.535	17.59	4	109.37	60.75	82.46	68.16
37........	0.373	0.531	17.53	4	115.25	60.18	81.62	68.34
36........	0.373	0.526	17.54	4	120.00	59.69	79.81	67.34
35........	0.373	0.520	17.55	4	125.00	59.00	76.98	65.67
34........	0.373	0.511	17.57	3	131.00	58.03	72.61	62.90

TABLE 78.

Holyoke Tests of a 30 1/2 Right Hand Special Smith Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
11........	1.000	1.006	17.19	4	136.50	50.65	75.31	76.26
10........	1.000	1.005	17.19	5	142.60	50.60	76.96	78.01
9........	1.000	1.006	17.23	3	157.00	50.70	81.91	82.67
8........	1.000	1.002	17.26	4	165.00	50.55	83.12	83.99
7........	1.000	1.007	17.18	4	173.50	50.65	84.28	85.39
6........	1.000	1.003	17.16	3	180.33	50.46	84.35	85.89
12........	1.000	1.003	17.21	4	189.25	50.51	85.12	86.33
5........	1.000	1.001	17.10	3	195.67	50.27	84.48	86.65
4........	1.000	0.998	17.11	3	213.00	50.13	84.30	86.66
3........	1.000	0.991	17.11	4	229.75	49.75	79.91	82.77
2........	1.000	0.973	17.16	4	243.75	48.91	73.09	76.78
1........	1.000	0.954	17.23	4	261.80	48.07	62.80	66.85
23........	0.900	0.939	17.35	4	137.50	47.47	73.39	78.56
22........	0.900	0.940	17.33	3	150.67	47.52	76.80	82.22
21........	0.900	0.938	17.30	4	159.00	47.38	78.19	84.10
20........	0.900	0.936	17.30	3	166.67	47.24	78.96	85.18
19........	0.900	0.933	17.28	4	174.75	47.06	79.64	86.35
18........	0.900	0.931	17.29	4	183.50	47.01	80.33	87.14
17........	0.900	0.928	17.30	5	192.40	46.87	80.77	87.82
16........	0.900	0.926	17.30	4	202.00	46.73	79.95	87.18
15........	0.900	0.918	17.31	4	210.25	46.37	78.17	85.86
14........	0.900	0.902	17.35	4	224.00	45.63	73.88	82.28
13........	0.900	0.878	17.38	3	256.33	44.44	61.49	70.19
25........	0.800	0.868	17.34	4	138.75	43.90	69.06	79.99
24........	0.800	0.869	17.36	3	150.00	43.95	71.96	83.16
26........	0.800	0.866	17.33	4	155.75	43.77	71.92	83.59
27........	0.800	0.861	17.33	3	166.00	43.50	73.66	86.15
28........	0.800	0.860	17.34	4	175.50	43.45	74.72	87.44
29........	0.800	0.856	17.33	4	183.50	43.27	74.83	87.98
30........	0.800	0.852	17.33	3	187.67	43.05	74.28	87.78
31........	0.800	0.849	17.38	4	193.25	42.96	74.17	87.58
32........	0.800	0.845	17.38	3	197.00	42.74	73.24	86.93
33........	0.800	0.837	17.37	3	205.00	42.34	71.30	85.48
34........	0.800	0.829	17.38	4	213.75	41.93	69.22	83.74
35........	0.800	0.819	17.43	3	227.67	41.53	65.53	79.82
36........	0.800	0.816	17.45	3	235.00	41.36	63.42	77.47
48........	0.700	0.790	17.46	4	139.75	40.07	64.53	81.32
47........	0.700	0.788	17.48	3	151.33	39.99	66.25	83.56
46........	0.700	0.785	17.48	3	161.00	39.85	67.58	85.54
44........	0.700	0.781	17.49	4	169.75	39.63	68.20	86.75
45........	0.700	0.778	17.48	4	174.25	39.50	67.92	86.73
43........	0.700	0.772	17.46	3	180.33	39.15	67.05	86.48
42........	0.700	0.768	17.44	3	183.67	38.93	66.09	85.82
41........	0.700	0.763	17.45	3	190.67	38.67	65.17	85.15
40........	0.700	0.754	17.47	3	199.00	38.28	63.25	83.38
39........	0.700	0.749	17.51	3	207.00	38.07	62.07	82.09
38........	0.700	0.740	17.61	3	221.67	37.72	59.82	79.40
37........	0.700	0.731	17.62	4	233.50	37.25	56.01	75.24

TABLE 78—Continued.

1	2	3	4	5	6	7	8	9
49........	0.600	0.693	17.55	3	132.00	35.24	55.41	78.99
50........	0.600	0.694	17.57	3	144.67	35.32	58.13	82.58
51........	0.600	0.691	17.59	3	155.00	35.20	59.49	84.71
52........	0.600	0.687	17.59	4	162.25	34.99	59.35	85.02
53........	0.600	0.681	17.59	4	169.50	34.69	58.95	85.18
54........	0.600	0.676	17.61	4	175.50	34.44	57.88	84.15
55........	0.600	0.665	17.68	4	187.25	33.94	56.15	82.49
62........	0.500	0.590	17.85	3	131.00	30.28	47.13	76.89
61........	0.500	0.590	17.86	4	140.00	30.28	48.69	79.39
63........	0.500	0.591	17.88	4	148.50	30.36	49.87	80.99
60........	0.500	0.590	17.85	4	151.50	30.24	49.97	81.62
59........	0.500	0.587	17.81	3	156.00	30.08	49.58	81.60
64........	0.500	0.584	17.89	4	162.75	29.96	49.77	81.88
58........	0.500	0.580	17.84	3	166.67	29.72	48.97	81.44
57........	0.500	0.572	17.85	4	177.50	29.32	47.90	80.69
56........	0.500	0.562	17.88	3	192.33	28.84	46.13	78.88

TABLE 79.

Holyoke Test of a 32-inch Left Hand Wellman, Seaver, Morgan Co. (Runner No 11) Turbine Wheel. Made Jan. 31, 1912. Swing-gate. Conical Draft-Tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
75........	Full	1.034	16.34	3	147.33	57.18	85.11	80.32
74........	Full	1.022	16.39	3	155.00	56.60	84.83	80.63
73........	Full	1.014	16.38	3	164.33	56.11	84.95	81.50
72........	Full	1.006	16.37	4	173.25	55.68	84.28	81.53
71........	Full	0.997	16.35	4	183.25	55.10	83.58	81.80
70........	Full	0.989	16.35	4	193.50	54.71	82.36	81.19
69........	Full	0.987	16.36	4	204.00	54.57	80.63	79.64
68........	Full	0.978	16.33	3	214.00	54.04	78.08	78.01
67........	Full	0.965	16.25	3	225.00	53.18	72.51	73.99
66........	Full	0.720	16.55	4	296.75	40.08	....	....
58........	0.862	0.941	15.82	4	140.25	51.20	75.05	81.70
57........	0.862	0.935	15.53	4	145.50	50.39	73.43	82.74
56........	0.862	0.926	15.61	5	150.00	50.02	72.97	82.40
59........	0.862	0.918	16.16	4	174.25	50.44	79.47	85.97
62........	0.862	0.915	16.13	4	180.50	50.25	79.57	86.57
60........	0.862	0.912	16.18	5	187.20	50.16	79.68	86.57
61........	0.862	0.907	16.16	4	199.25	49.83	78.75	86.24
63........	0.862	0.898	16.14	3	208.67	49.31	76.13	84.35
64........	0.862	0.876	16.88	4	225.50	49.22	72.67	77.13
65........	0.862	0.650	16.89	4	299.75	36.53	....	....
50........	0.755	0.824	16.39	4	159.25	45.62	72.63	85.65
51........	0.755	0.823	16.40	4	169.75	45.57	74.32	87.69
52........	0.755	0.822	16.39	4	175.25	45.53	74.60	88.14
53........	0.755	0.820	16.38	4	181.75	45.39	75.15	89.13
54........	0.755	0.818	16.36	3	187.00	45.26	75.05	89.37
55........	0.755	0.812	16.32	4	191.25	44.84	73.27	88.28
40........	0.734	0.823	16.62	4	136.00	45.89	70.29	81.27
39........	0.734	0.816	16.64	4	145.75	45.53	71.79	83.55
38........	0.734	0.810	16.65	4	153.50	45.21	72.80	85.28
37........	0.734	0.804	16.66	4	160.25	44.89	73.08	86.17
41........	0.734	0.804	16.62	4	171.00	44.84	74.87	88.58
42........	0.734	0.802	16.62	4	176.75	44.71	75.23	89.28
43........	0.734	0.800	16.61	4	182.00	44.57	75.26	89.63
44........	0.734	0.793	16.58	4	187.50	44.17	74.11	89.23
45........	0.734	0.781	16.57	3	196.00	43.50	71.51	87.48
46........	0.734	0.768	16.58	3	203.33	42.78	68.00	84.54
47........	0.734	0.756	16.58	3	211.67	42.07	64.36	81.35
48........	0.734	0.731	16.63	4	229.75	40.74	55.88	72.73
49........	0.734	0.568	16.89	4	293.50	31.90	....	....
25........	0.669	0.752	16.78	4	131.00	42.15	63.73	79.45
24........	0.669	0.744	16.78	4	146.00	41.67	66.58	83.97
27........	0.669	0.746	16.92	4	152.50	41.98	68.62	85.19
26........	0.669	0.745	16.95	4	157.50	41.93	69.43	86.15
23........	0.669	0.734	16.78	4	159.50	41.14	67.89	86.72
31........	0.669	0.741	16.94	4	166.75	41.71	70.47	87.94
28........	0.669	0.736	16.92	4	172.75	41.40	70.38	88.59
29........	0.669	0.729	16.91	4	178.50	41.00	69.47	88.35
30........	0.669	0.721	16.92	4	185.75	40.56	67.77	87.07
32........	0.669	0.706	17.08	4	197.25	39.90	65.97	85.35
33........	0.669	0.694	17.12	4	207.75	39.29	63.16	82.80
34........	0.669	0.681	17.13	4	219.25	38.55	59.99	80.11
35........	0.669	0.670	17.14	4	228.50	37.95	55.58	75.34
36........	0.669	0.525	17.33	4	291.75	29.88	....	....

TABLE 79—Continued.

1	2	3	4	5	6	7	8	9
15........	0.518	0.567	17.08	4	127.50	32.06	48.07	77.40
14........	0.518	0.570	17.07	4	145.50	32.19	51.32	82.35
13........	0.518	0.565	17.11	4	154.25	31.98	51.59	83.13
22........	0.518	0.566	17.01	4	153.25	31.90	51.25	83.29
21........	0.518	0.554	17.03	4	167.50	31.25	50.93	84.38
20........	0.518	0.539	17.07	4	181.00	30.44	49.53	84.05
16........	0.518	0.523	17.14	4	199.25	29.60	47.25	82.12
19........	0.518	0.513	17.12	3	210.67	29.04	44.84	79.52
17........	0.518	0.500	17.19	4	221.50	28.33	40.41	73.16
18........	0.518	0.486	17.19	3	230.67	27.58	35.07	65.22
7........	0.518	0.401	17.48	4	272.50	22.95	....	....
6........	0.296	0.304	17.73	4	107.50	17.53	24.84	70.47
5........	0.296	0.300	17.75	4	120.00	17.30	25.54	73.34
4........	0.296	0.293	17.80	5	141.40	16.93	25.79	75.47
3........	0.296	0.283	17.80	4	154.00	16.32	23.41	71.06
2........	0.296	0.271	17.84	4	181.50	15.66	22.07	69.67
1........	0.296	0.230	17.96	5	259.00	13.31	....	....

TABLE 80.

Holyoke Tests of a 30-inch Right Hand Allis-Chalmers (Runner No. 18) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of opening of the speed-gate. In inches.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
80........	2.75	1.010	17.49	4	130.00	64.45	91.97	71.94
79........	2.75	1.009	17.56	4	148.00	64.52	98.44	76.61
78........	2.75	1.005	17.59	4	157.50	64.32	99.99	77.93
77........	2.75	1.002	17.56	4	166.00	64.04	100.37	78.70
76........	2.75	1.002	17.47	4	176.50	63.91	101.38	80.07
75........	2.75	1.000	17.44	4	186.25	63.70	101.35	80.45
74........	2.75	1.000	17.39	4	196.00	63.64	100.73	80.26
73........	2.75	0.991	17.39	4	204.25	63.02	98.80	79.49
81........	2.75	0.982	17.50	4	213.00	62.68	96.59	77.65
72........	2.75	0.982	17.44	4	211.25	62.55	95.80	77.43
71........	2.75	0.974	17.48	4	220.25	62.14	93.22	75.67
70........	2.75	0.967	17.50	4	229.50	61.74	90.20	73.61
69........	2.75	0.964	17.50	4	239.00	61.53	86.71	71.00
15........	2.50	0.965	17.20	4	132.50	61.06	89.73	75.33
14........	2.50	0.967	17.14	4	141.50	61.06	92.40	77.85
13........	2.50	0.961	17.16	4	154.00	60.73	94.98	80.36
12........	2.50	0.959	17.18	3	165.00	60.66	96.77	81.88
10........	2.50	0.959	17.20	5	176.60	60.66	98.24	83.02
11........	2.50	0.957	17.18	4	181.00	60.52	98.50	83.53
9........	2.50	0.959	17.23	4	187.25	60.73	99.63	83.96
16........	2.50	0.959	17.22	5	187.20	60.73	99.61	83.98
8........	2.50	0.954	17.25	3	193.00	60.46	99.19	83.86
7........	2.50	0.951	17.23	4	197.25	60.19	97.80	83.15
6........	2.50	0.946	17.23	4	202.50	59.92	96.73	82.61
5........	2.50	0.942	17.18	5	208.20	59.59	94.41	81.32
4........	2.50	0.936	17.22	4	218.25	59.25	92.37	79.83
3........	2.50	0.931	17.22	5	228.20	58.92	89.69	77.94
2........	2.50	0.925	17.24	5	238.40	58.59	86.49	75.50
1........	2.50	0.804	17.42	4	337.25	51.17	....	....
29........	2.125	0.877	17.44	3	134.00	55.88	85.88	77.71
28........	2.125	0.874	17.45	3	144.00	55.68	88.81	80.60
27........	2.125	0.876	17.44	3	156.00	55.81	92.44	83.74
25........	2.125	0.875	17.44	4	170.00	55.75	95.59	86.69
24........	2.125	0.872	17.45	4	179.00	55.55	96.32	87.62
23........	2.125	0.869	17.45	3	185.00	55.36	96.20	87.81
26........	2.125	0.867	17.45	3	189.00	55.22	95.99	87.84
22........	2.125	0.864	17.45	4	190.00	55.09	95.35	87.46
21........	2.125	0.860	17.45	3	198.00	54.83	94.58	87.16
20........	2.125	0.857	17.44	3	206.00	54.57	93.42	86.55
19........	2.125	0.853	17.43	4	216.75	54.31	91.74	85.45
18........	2.125	0.847	17.41	3	226.00	53.92	88.82	83.43
17........	2.125	0.838	17.44	4	233.75	53.41	84.80	80.28
40........	1.75	0.780	17.56	4	135.75	49.84	77.98	78.56
41........	1.75	0.782	17.57	4	146.00	50.03	81.66	81.91
39........	1.75	0.783	17.55	4	156.50	50.03	85.16	85.53
43........	1.75	0.780	17.59	4	161.00	49.90	85.67	86.06
38........	1.75	0.777	17.57	4	168.75	49.71	86.73	87.56
37........	1.75	0.775	17.56	4	175.50	49.53	87.01	88.22
36........	1.75	0.770	17.55	4	181.50	49.21	86.70	88.52
35........	1.75	0.768	17.54	3	188.00	49.09	86.39	88.47
34........	1.75	0.767	17.48	3	193.33	48.90	85.33	88.03
33........	1.75	0.759	17.47	4	203.75	48.40	83.77	87.36
32........	1.75	0.750	17.49	4	212.50	47.84	80.95	85.30
42........	1.75	0.747	17.63	4	221.75	47.84	80.45	84.10
31........	1.75	0.736	17.49	5	230.50	46.96	76.65	82.29
30........	1.75	0.726	17.53	4	238.00	46.34	71.95	78.10

TABLE 80—Continued.

1	2	3	4	5	6	7	8	9
54........	1.50	0.684	17.70	3	120.67	43.90	64.21	72.86
53........	1.50	0.697	17.70	4	145.75	44.75	73.14	81.43
52........	1.50	0.695	17.70	4	162.40	44.63	76.59	85.49
51........	1.50	0.691	17.69	4	173.50	44.33	77.63	87.29
50........	1.50	0.688	17.70	3	181.67	44.14	77.99	88.02
55........	1.50	0.687	17.71	4	181.75	44.08	78.02	88.13
49........	1.50	0.683	17.68	3	187.00	43.78	76.89	87.59
56........	1.50	0.680	17.71	4	189.75	43.66	76.87	87.66
48........	1.50	0.677	17.70	4	193.50	43.48	76.05	87.13
47........	1.50	0.671	17.71	3	204.67	43.05	74.25	85.87
46........	1.50	0.660	17.71	3	215.33	42.39	71.61	84.11
45........	1.50	0.650	17.74	4	225.00	41.74	68.02	81.00
44........	1.50	0.639	17.75	4	234.50	41.08	63.80	77.16
64........	1.125	0.544	18.13	4	127.00	35.35	52.98	72.90
63........	1.125	0.549	18.14	3	145.00	35.68	56.99	77.64
66........	1.125	0.552	18.23	4	156.75	35.97	59.24	79.65
62........	1.125	0.555	18.15	4	167.25	36.08	60.68	81.70
65........	1.125	0.552	18.18	4	174.50	35.91	60.67	81.94
61........	1.125	0.547	18.11	4	180.50	35.51	60.03	82.30
68........	1.125	0.546	18.12	4	181.25	35.46	60.28	82.72
67........	1.125	0.538	18.23	5	189.40	35.06	60.12	82.94
60........	1.125	0.535	18.12	4	194.75	34.73	58.88	82.50
59........	1.125	0.523	18.11	4	208.25	33.94	56.66	81.29
58........	1.125	0.513	18.08	5	222.40	33.28	53.79	78.82
57........	1.125	0.504	18.04	4	234.25	32.67	49.57	74.17

TABLE 81.

Holyoke Tests of a 31-inch Right Hand Wellman-Seaver-Morgan Co. Turbine. Wheel supported by Ball-bearing step. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
49........	1.000	1.049	17.15	3	134.67	79.75	115.81	74.66
48........	1.000	1.038	17.15	3	147.00	78.95	119.29	77.68
47........	1.000	1.025	17.19	3	165.67	78.02	123.40	81.13
46........	1.000	1.024	17.18	2	174.00	77.91	124.34	81.91
45........	1.000	1.018	17.19	3	178.33	77.52	124.19	82.18
44........	1.000	1.013	17.19	3	183.33	77.16	124.35	82.66
43........	1.000	0.012	17.19	4	186.25	77.09	124.07	82.56
42........	1.000	1.009	17.18	4	189.50	76.82	123.94	82.81
41........	1.000	1.007	17.19	3	193.00	76.67	123.89	82.89
40........	1.000	1.006	17.12	4	195.50	76.44	123.13	82.96
39........	1.000	1.002	17.09	4	200.25	76.10	122.48	83.04
38........	1.000	0.999	17.10	4	206.25	75.87	122.41	83.19
37........	1.000	0.997	17.07	3	210.33	75.66	121.01	82.62
36........	1.000	0.993	17.09	4	219.00	75.38	119.36	81.70
35........	1.000	0.906	17.24	4	258.75	69.11	78.35	57.98
34........	1.000	0.744	17.45	4	302.25	57.07	....	....
32........	0.883	0.909	17.18	3	129.33	69.23	101.03	74.90
31........	0.883	0.904	17.20	4	154.25	68.82	111.16	82.81
33........	0.883	0.901	17.19	4	165.00	68.60	113.91	85.18
30........	0.883	0.896	17.20	3	172.33	68.25	114.80	86.23
29........	0.883	0.892	17.21	4	181.50	67.97	116.51	87.82
27........	0.883	0.888	17.25	3	188.00	67.77	117.27	88.45
26........	0.883	0.886	17.30	4	194.00	67.71	117.49	88.44
28........	0.883	0.885	17.23	4	197.25	67.43	117.06	88.85
25........	0.883	0.883	17.31	4	201.75	67.49	116.07	87.61
24........	0.883	0.857	17.28	4	213.25	65.42	109.77	85.62
23........	0.883	0.818	17.31	4	227.00	62.53	96.23	78.39
22........	0.883	0.761	17.42	4	244.25	58.32	73.96	64.19
21........	0.883	0.620	17.65	4	291.75	47.83	....	....
69........	0.750	0.835	17.28	4	124.75	63.73	91.41	73.19
68........	0.750	0.839	17.23	4	148.75	63.93	103.59	82.93
67........	0.750	0.834	17.23	4	168.75	63.60	109.35	87.99
64........	0.750	0.830	17.24	4	179.25	63.26	110.72	89.52
65........	0.750	0.829	17.23	4	182.00	63.20	110.77	89.69
63........	0.750	0.828	17.25	4	186.25	63.12	111.66	90.43
66........	0.750	0.824	17.23	3	188.67	62.85	110.83	90.24
62........	0.750	0.821	17.29	4	191.75	62.72	110.32	89.70
61........	0.750	0.809	17.30	4	197.50	61.81	107.64	88.76
60........	0.750	0.796	17.34	4	203.25	60.86	104.62	87.42
59........	0.750	0.768	17.38	4	212.50	58.80	96.52	83.28
58........	0.750	0.693	17.55	4	237.00	53.31	71.76	67.64
57........	0.750	0.573	17.75	4	287.00	44.32	....	....
20........	0.667	0.726	17.27	4	106.25	55.43	70.78	65.20
19........	0.667	0.737	17.26	3	148.67	56.25	90.93	82.59
18........	0.667	0.739	17.25	4	161.00	56.39	95.55	86.62
17........	0.667	0.738	17.24	4	168.50	56.31	96.94	88.05
16........	0.667	0.733	17.27	3	173.67	55.92	96.76	88.35
15........	0.667	0.722	17.30	4	179.25	55.17	95.52	88.25
14........	0.667	0.699	17.34	4	189.50	53.48	91.81	87.30
13........	0.667	0.671	17.39	4	201.75	51.37	85.52	84.42
11........	0.667	0.508	17.63	4	280.75	39.17	....	....

TABLE 81—Continued.

1	2	3	4	5	6	7	8	9
8........	0.500	0.554	17.82	3	117.00	42.95	60.23	69.39
7........	0.500	0.546	17.91	3	135.00	42.46	65.40	75.84
10........	0.500	0.548	17.67	3	151.00	42.28	68.58	80.95
6........	0.500	0.548	18.05	4	157.50	42.72	71.54	81.80
9........	0.500	0.547	17.71	4	157.00	42.28	69.41	81.73
5........	0.500	0.539	18.12	5	167.60	42.11	71.05	82.10
4........	0.500	0.512	18.13	4	187.00	40.02	67.95	82.58
3........	0.500	0.488	18.18	4	213.00	38.21	58.05	73.68
2........	0.500	0.460	18.07	4	232.00	35.92	42.15	57.26
1........	0.500	0.402	18.20	3	275.00	31.50	....	....
52........	0.333	0.362	18.18	3	96.00	28.35	34.88	59.68
55........	0.333	0.361	18.01	3	117.33	28.10	39.08	68.09
51........	0.333	0.348	18.19	4	133.00	27.25	40.27	71.64
54........	0.333	0.347	18.21	3	139.67	27.20	40.60	72.28
53........	0.333	0.340	18.22	3	143.67	26.62	39.15	71.18
56........	0.333	0.333	18.06	4	148.00	26.00	37.64	70.69
50........	0.333	0.316	18.26	3	201.67	24.82	36.64	71.28

TABLE 82.

Holyoke Tests of a 30-inch Right Hand Smith (Type K) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
21........	1.000	0.859	15.60	4	149.50	74.08	106.48	81.24
20........	1.000	0.851	15.65	4	164.50	73.50	109.22	83.72
19........	1.000	0.842	15.68	3	183.67	72.79	110.86	85.65
18........	1.000	0.837	15.70	4	204.50	72.36	111.09	86.22
17........	1.000	0.815	15.74	4	225.25	70.52	101.97	81.00
16........	1.000	0.744	15.88	4	244.25	64.70	73.71	63.26
15........	1.000	0.613	16.19	4	298.75	53.82	....	....
27........	0.928	0.793	15.59	3	126.00	68.35	91.26	75.52
26........	0.928	0.794	15.60	3	142.00	68.42	98.57	81.43
25........	0.928	0.790	15.67	4	154.25	68.21	102.41	84.49
23........	0.928	0.784	15.69	4	162.00	67.80	103.65	85.91
24........	0.928	0.783	15.68	4	166.00	67.66	104.20	86.61
22........	0.928	0.780	15.70	4	174.75	67.45	105.48	87.83
28........	0.928	0.777	15.58	4	183.25	66.90	105.08	88.89
29........	0.928	0.773	15.56	3	194.33	66.55	105.57	89.89
30........	0.928	0.750	15.58	3	204.00	64.63	98.51	86.26
31........	0.928	0.667	15.70	4	232.75	57.69	70.24	68.38
32........	0.928	0.554	15.92	3	291.33	48.23	....	....
77........	0.838	0.757	15.71	3	114.67	65.52	83.06	71.15
76........	0.838	0.759	15.78	4	128.50	65.80	89.97	76.40
75........	0.838	0.759	15.77	4	144.75	65.80	96.98	82.41
74........	0.838	0.759	15.65	6	153.83	65.52	99.35	85.43
73........	0.838	0.758	15.64	3	160.00	65.38	100.44	86.61
78........	0.838	0.755	15.60	4	164.25	65.11	101.12	87.79
72........	0.838	0.753	15.65	4	168.50	65.04	101.70	88.10
79........	0.838	0.751	15.49	4	172.50	64.49	101.00	89.15
80........	0.838	0.747	15.54	3	184.33	64.29	102.36	90.34
81........	0.838	0.719	15.64	3	198.00	62.04	95.61	86.88
82........	0.838	0.647	15.78	3	230.33	56.11	69.51	69.23
83........	0.838	0.546	16.01	4	291.50	47.67	....	....
98........	0.752	0.691	15.52	3	140.33	59.42	84.70	80.99
99........	0.752	0.693	15.51	4	149.00	59.55	88.14	84.14
94........	0.752	0.699	15.55	4	157.75	60.15	92.36	87.07
93........	0.752	0.698	15.55	4	164.75	60.09	93.47	88.21
95........	0.752	0.697	15.55	5	166.40	59.95	92.40	87.40
92........	0.752	0.695	15.55	3	172.00	59.82	93.44	88.57
96........	0.752	0.687	15.52	3	174.17	59.09	92.51	88.95
97........	0.752	0.681	15.51	3	177.67	58.49	91.15	88.60
34........	0.752	0.688	16.12	4	171.25	60.29	98.20	89.09
33........	0.752	0.679	16.14	4	178.50	59.55	96.97	88.96
39........	0.714	0.640	16.16	4	128.00	56.17	78.03	75.80
38........	0.714	0.644	16.16	4	138.25	56.50	81.78	78.98
35........	0.714	0.656	16.15	3	158.00	57.56	90.60	85.92
37........	0.714	0.649	16.15	4	160.25	56.90	89.95	86.32
43........	0.714	0.647	16.11	4	164.25	56.63	90.71	87.68
36........	0.714	0.651	16.15	5	168.40	57.10	91.48	87.47
40........	0.714	0.644	16.14	4	169.50	56.44	90.03	87.15
41........	0.714	0.636	16.13	5	172.20	55.78	88.35	86.58
42........	0.714	0.623	16.15	4	177.50	54.67	85.71	85.60
44........	0.714	0.568	16.23	4	206.25	49.93	74.69	81.28
45........	0.714	0.535	16.31	4	236.75	47.17	57.16	65.51
46........	0.714	0.456	16.45	4	287.75	40.39	....	....

TABLE 82—Continued.

1	2	3	4	5	6	7	8	9
64........	0.656	0.609	15.97	2	106.50	53.11	64.28	66.83
63........	0.656	0.611	15.96	3	145.00	53.24	78.77	81.74
62........	0.656	0.614	15.96	4	160.25	53.50	81.25	83.90
60........	0.656	0.602	16.01	4	166.25	52.54	80.28	84.15
61........	0.656	0.599	16.01	4	168.00	52.28	79.60	83.86
59........	0.656	0.590	16.04	4	173.00	51.58	80.40	85.69
58........	0.656	0.583	16.07	3	176.33	51.00	79.82	85.88
65........	0.656	0.569	16.02	3	182.67	49.68	77.18	85.51
66........	0.656	0.544	16.07	2	199.50	47.61	72.25	83.27
67........	0.656	0.438	16.29	3	284.33	38.57	....	....
91........	0.536	0.495	15.93	3	124.33	43.13	57.78	74.16
90........	0.536	0.494	15.92	3	138.33	43.01	60.95	78.49
89........	0.536	0.493	15.95	4	149.00	43.01	62.95	80.92
86........	0.536	0.484	16.01	3	159.00	42.29	62.38	81.24
87........	0.536	0.473	16.01	4	169.00	41.28	61.20	81.66
88........	0.536	0.458	16.02	4	180.50	39.98	59.92	82.49
85........	0.536	0.455	16.09	4	191.75	39.80	57.87	79.68
84........	0.536	0.367	16.29	5	272.80	32.30	....	....
53........	0.409	0.384	16.51	4	97.75	34.01	39.53	62.08
52........	0.409	0.376	16.54	3	127.67	33.40	45.47	72.57
51........	0.409	0.372	16.55	3	139.67	33.07	46.37	74.70
50........	0.409	0.364	16.58	3	147.67	32.35	45.46	74.73
49........	0.409	0.358	16.60	4	154.00	31.86	44.62	74.39
48........	0.409	0.354	16.62	4	161.50	31.48	43.87	73.93
54........	0.409	0.349	16.58	4	170.50	31.04	42.19	72.29
47........	0.409	0.347	16.65	4	174.25	30.93	42.07	72.03
55........	0.409	0.340	16.60	4	186.75	30.23	39.45	69.32
56........	0.409	0.320	16.65	3	221.67	28.46	26.78	49.79
57........	0.409	0.290	16.68	3	258.67	25.87	....	....
70........	0.104	0.102	17.01	3	158.67	9.15	....	....
71........	0.138	0.121	16.97	3	182.00	10.86	....	....
69........	0.171	0.142	16.90	3	201.67	12.77	....	....
68........	0.226	0.189	16.80	3	223.00	16.86	....	....

TABLE 83.

Holyoke Tests of a 28-inch Right Hand Wellman-Seaver-Morgan Co. (Runner No. 40) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of opening of the speed-gate. In inches.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
9........	3.20	1.018	17.20	4	134.00	79.68	113.22	72.84
8........	3.20	1.013	17.25	5	156.80	79.39	123.02	79.21
7........	3.20	1.000	17.16	4	171.50	78.15	124.20	81.66
6........	3.20	0.983	17.20	4	181.75	76.91	120.65	80.42
5........	3.20	0.970	17.23	4	196.50	75.98	118.59	79.87
4........	3.20	0.959	17.25	4	214.75	75.11	116.64	79.38
3........	3.20	0.953	17.30	4	235.25	74.75	113.58	77.44
2........	3.20	0.948	17.31	4	257.75	74.39	108.89	74.56
1........	3.20	0.944	17.32	4	281.00	74.11	101.75	69.90
17........	2.80	0.929	17.24	4	146.75	72.74	113.36	79.71
16........	2.80	0.917	17.28	4	160.50	71.89	116.23	82.50
15........	2.80	0.901	17.36	4	174.25	70.83	115.67	82.95
14........	2.80	0.890	17.43	4	193.50	70.13	116.78	84.24
13........	2.80	0.887	17.45	4	215.75	69.92	117.19	84.69
12........	2.80	0.884	17.45	4	240.50	69.64	116.11	84.25
11........	2.80	0.878	17.47	4	262.00	69.22	110.68	80.70
10........	2.80	0.858	17.49	4	274.00	67.69	99.21	73.89
26........	2.40	0.801	17.48	4	144.25	63.16	98.37	78.57
25........	2.40	0.792	17.55	4	163.75	62.62	103.76	83.25
24........	2.40	0.788	17.51	4	183.00	62.21	107.13	86.72
22........	2.40	0.784	17.52	4	202.50	61.94	109.99	89.37
27........	2.40	0.786	17.53	4	213.75	62.07	110.94	89.90
23........	2.40	0.786	17.52	5	215.40	62.07	110.49	89.59
28........	2.40	0.786	17.54	5	222.60	62.07	111.50	90.31
21........	2.40	0.784	17.52	4	228.25	61.87	110.20	89.64
20........	2.40	0.769	17.52	4	244.00	60.73	103.08	85.42
19........	2.40	0.750	17.56	4	260.25	59.32	94.24	79.77
18........	2.40	0.729	17.60	4	275.25	57.66	83.06	72.17
45........	2.18	0.749	17.64	4	195.50	59.32	106.19	89.48
50........	2.18	0.749	17.69	4	203.50	59.45	107.46	90.10
44........	2.18	0.749	17.64	5	209.80	59.32	107.62	90.69
49........	2.18	0.750	17.66	4	211.25	59.45	108.37	91.01
46........	2.18	0.749	17.60	4	215.50	59.25	107.94	91.27
43........	2.18	0.747	17.63	4	221.50	59.19	106.94	90.36
47........	2.18	0.745	17.58	4	225.00	58.92	105.91	90.16
48........	2.18	0.739	17.58	5	228.40	58.45	103.38	88.71
42........	2.18	0.739	17.67	4	229.25	58.59	103.76	88.38
41........	2.18	0.732	17.70	4	239.00	58.12	100.96	86.54
39........	2.00	0.700	17.70	4	143.50	55.55	86.60	77.66
34........	2.00	0.701	17.61	4	157.50	55.49	90.30	81.48
32........	2.00	0.701	17.63	4	175.75	55.49	95.46	86.04
31........	2.00	0.700	17.63	4	193.50	55.42	99.26	89.58
40........	2.00	0.699	17.68	6	202.33	55.42	100.74	90.65
30........	2.00	0.696	17.65	4	206.75	55.16	99.82	90.40
33........	2.00	0.688	17.62	4	214.50	54.51	97.09	89.13
35........	2.00	0.689	17.64	6	214.86	54.57	97.25	89.08
29........	2.00	0.683	17.70	4	226.25	54.18	95.58	87.88
36........	2.00	0.667	17.74	5	246.00	52.96	89.08	83.60
37........	2.00	0.648	17.77	4	265.25	51.55	80.04	77.04
38........	2.00	0.633	17.81	4	283.75	50.40	68.50	67.29

TABLE 83—Continued.

1	2	3	4	5	6	7	8	9
58........	1.60	0.549	18.05	4	149.25	44.02	70.26	77.97
57........	1.60	0.552	18.06	5	168.40	44.26	74.19	81.84
56........	1.60	0.550	18.10	3	184.00	44.14	75.51	83.34
55........	1.60	0.543	18.13	4	200.00	43.60	76.04	84.82
54........	1.60	0.532	18.23	4	216.00	42.87	75.61	85.30
53........	1.60	0.522	18.25	4	237.25	42.03	71.59	82.30
52........	1.60	0.507	18.26	4	261.00	40.84	63.00	74.50
51........	1.60	0.497	18.20	4	282.50	39.96	51.15	62.01
65........	1.18	0.399	18.30	4	154.00	32.18	51.12	76.54
64........	1.18	0.395	18.30	4	168.00	31.90	50.69	76.57
63........	1.18	0.392	18.32	4	184.50	31.69	50.10	76.10
62........	1.18	0.386	18.33	4	203.50	31.19	49.12	75.77
61........	1.18	0.378	18.35	4	219.25	30.54	46.31	72.87
60........	1.18	0.374	18.38	4	233.75	30.27	42.33	67.07
59........	1.18	0.360	18.40	4	261.00	29.14	31.50	51.81
70........	0.88	0.290	18.51	3	158.33	23.50	35.35	71.67
69........	0.88	0.284	18.62	4	180.00	23.15	33.67	68.89
68........	0.88	0.280	18.65	3	198.00	22.81	29.87	61.92
66........	0.88	0.276	18.59	4	217.75	22.47	26.28	55.48
67........	0.88	0.268	18.64	4	255.25	21.79	15.40	33.44
73........	3.20	0.821	17.64	4	358.25	65.07	....	....
72........	2.80	0.740	17.75	4	365.25	58.79	....	....
71........	2.40	0.643	17.84	4	360.00	51.23	....	....
74........	2.18	0.590	17.92	4	354.25	47.09	....	....
75........	2.00	0.550	17.98	4	350.75	43.96	....	....
76........	1.60	0.442	18.12	4	332.00	35.46	....	....
77........	1.18	0.333	18.36	4	306.75	26.94	....	....
78........	0.88	0.258	18.47	4	281.50	20.92	....	....

TABLE 84.

Holyoke Tests of a 26.81-inch Right Hand I. P. Morris (Runner "M") Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of opening of the speed-gate. In inches.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
56........	4.00	1.013	17.36	4	159.00	66.17	96.26	73.89
55........	4.00	1.010	17.38	4	189.00	65.97	101.83	78.31
53........	4.00	1.003	17.40	4	200.50	65.55	101.96	78.83
52........	4.00	1.001	17.39	4	211.00	65.42	102.19	79.20
51........	4.00	0.998	17.43	5	222.00	65.28	102.14	79.15
54........	4.00	0.998	17.41	5	225.00	65.28	102.16	79.26
50........	4.00	0.999	17.42	4	232.00	65.35	102.53	79.41
48........	4.00	1.000	17.35	4	239.00	65.28	101.28	78.85
49........	4.00	1.000	17.41	4	250.25	65.42	101.51	78.58
47........	4.00	1.004	17.33	4	272.25	65.48	98.89	76.84
46........	4.00	1.001	17.22	3	301.00	65.07	91.11	71.70
45........	4.00	0.984	17.24	5	323.60	64.04	78.36	62.58
44........	4.00	0.947	17.32	4	340.00	61.74	61.75	50.92
67........	3.50	0.921	17.56	2	149.00	60.52	90.20	74.84
66........	3.50	0.924	17.56	4	177.25	60.66	96.58	79.94
65........	3.50	0.919	17.59	4	197.00	60.39	98.99	82.17
64........	3.50	0.918	17.62	4	212.50	60.39	100.35	83.15
63........	3.50	0.918	17.61	4	226.00	60.39	101.25	83.94
61........	3.50	0.918	17.58	4	229.75	60.32	101.54	84.43
60........	3.50	0.920	17.58	4	240.50	60.46	101.92	84.55
62........	3.50	0.921	17.58	4	247.50	60.52	101.89	84.44
59........	3.50	0.922	17.57	4	258.25	60.59	101.62	84.17
58........	3.50	0.911	17.56	3	299.00	59.85	90.51	75.93
57........	3.50	0.840	17.67	4	333.50	55.36	60.57	54.60
21........	3.00	0.823	17.37	3	160.33	53.79	82.50	77.86
19........	3.00	0.827	17.35	4	182.00	53.99	88.15	82.97
17........	3.00	0.830	17.32	3	212.67	54.11	92.70	87.21
18........	3.00	0.829	17.33	4	223.50	54.11	93.36	87.79
20........	3.00	0.829	17.40	3	232.00	54.18	94.10	88.01
16........	3.00	0.832	17.32	4	240.25	54.24	94.54	88.73
22........	3.00	0.833	17.35	3	245.00	54.38	94.93	88.71
15........	3.00	0.816	17.34	4	272.25	53.28	87.36	83.37
14........	3.00	0.783	17.40	4	299.75	51.17	72.59	71.88
72........	2.65	0.760	17.37	4	170.00	49.65	79.25	81.02
71........	2.65	0.769	17.30	3	204.33	50.15	86.59	88.00
73........	2.65	0.770	17.41	4	214.75	50.34	89.06	89.60
70........	2.65	0.769	17.22	4	215.25	50.03	87.31	89.36
74........	2.65	0.772	17.45	4	223.75	50.53	90.08	90.08
83........	2.65	0.772	17.48	4	230.25	50.59	90.61	90.34
69........	2.65	0.768	17.24	4	220.25	49.96	86.67	88.72
68........	2.65	0.764	17.04	4	215.25	49.40	84.70	88.72
77........	2.65	0.773	17.47	3	232.00	50.66	90.59	90.25
75........	2.65	0.772	17.44	4	233.75	50.53	90.57	90.62
86........	2.65	0.772	17.47	5	233.20	50.59	90.35	90.14
82........	2.65	0.770	17.47	4	234.25	50.47	90.05	90.05
85........	2.65	0.770	17.46	4	236.00	50.40	90.01	90.19
76........	2.65	0.770	17.45	6	235.67	50.40	89.89	90.11
81........	2.65	0.769	17.48	3	237.00	50.40	89.68	89.75
78........	2.65	0.768	17.47	4	238.50	50.34	89.52	89.75
79........	2.65	0.767	17.48	3	240.67	50.28	88.88	89.16
80........	2.65	0.765	17.48	4	243.00	50.15	88.27	88.78
84........	2.65	0.748	17.50	4	269.25	49.02	81.50	83.77

TABLE 84—Continued.

1	2	3	4	5	6	7	8	9
13........	2.50	0.712	17.51	3	139.00	46.71	67.32	72.57
11........	2.50	0.718	17.49	4	164.75	47.09	74.80	80.08
7........	2.50	0.727	17.51	3	201.00	47.65	82.75	87.44
12........	2.50	0.728	17.47	3	211.00	47.71	84.31	89.19
10........	2.50	0.729	17.47	5	216.60	47.77	85.23	90.05
8........	2.50	0.728	17.48	4	219.50	47.71	85.05	89.92
6........	2.50	0.728	17.47	3	222.00	47.71	84.67	89.57
9........	2.50	0.726	17.49	4	223.75	47.59	84.66	89.68
5........	2.50	0.726	17.48	4	225.25	47.59	84.55	89.61
4........	2.50	0.716	17.51	4	237.75	46.96	82.04	87.97
3........	2.50	0.704	17.52	4	256.00	46.16	77.49	84.49
2........	2.50	0.682	17.55	6	284.00	44.81	68.77	77.11
1........	2.50	0.658	17.59	4	311.00	43.23	56.48	65.49
32........	2.00	0.582	17.77	3	139.33	38.44	56.52	72.95
30........	2.00	0.587	17.74	4	166.00	38.73	62.31	79.96
29........	2.00	0.591	17.72	3	194.33	38.96	67.06	85.65
28........	2.00	0.582	17.73	3	208.33	38.38	65.58	84.98
26........	2.00	0.572	17.74	3	222.00	37.75	64.51	84.94
31........	2.00	0.570	17.76	3	226.67	37.63	64.50	85.09
27........	2.00	0.569	17.75	3	233.33	37.57	64.27	84.98
25........	2.00	0.565	17.75	3	243.00	37.34	63.26	84.15
33........	2.00	0.565	17.78	4	256.00	37.34	61.99	82.33
24........	2.00	0.550	17.76	4	272.25	36.31	57.69	78.88
23........	2.00	0.521	17.81	4	296.50	34.44	44.88	64.51
43........	1.25	0.367	18.21	3	145.67	24.54	37.05	73.08
42........	1.25	0.366	18.23	4	158.25	24.49	38.32	75.68
41........	1.25	0.363	18.26	3	168.00	24.34	38.65	76.67
40........	1.25	0.357	18.27	3	182.00	23.89	37.46	75.68
39........	1.25	0.347	18.27	4	205.50	23.25	36.08	74.89
38........	1.25	0.343	18.27	4	217.75	22.96	35.59	74.81
36........	1.25	0.339	18.32	4	227.75	22.71	34.47	73.05
37........	1.25	0.337	18.30	4	236.50	22.57	32.93	70.30
35........	1.25	0.331	18.37	5	250.20	22.23	30.29	65.41
34........	1.25	0.320	18.43	4	289.25	21.50	17.51	38.97
92........	4.00	0.824	17.43	3	382.00	53.92	....	....
91........	3.50	0.706	17.67	3	375.67	46.53	....	....
90........	3.00	0.610	17.78	4	367.50	40.31	....	....
89........	2.50	0.522	17.95	3	359.00	34.67	....	....
88........	2.00	0.439	18.11	4	348.75	29.25	....	....
87........	1.25	0.300	18.25	4	317.33	20.06	....	....

TABLE 85.

Holyoke Tests of a 51-inch Right Hand Special Victor Turbine. Swing-gate. Conical Draft-Tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
50........	1.000	0.979	15.52	4	77.00	184.51	227.82	70.24
49........	1.000	0.989	15.49	4	89.75	186.10	247.44	75.78
48........	1.000	0.992	15.52	3	99.33	186.97	260.49	79.25
47........	1.000	0.993	15.52	3	104.80	187.11	264.27	80.34
46........	1.000	0.997	15.49	4	110.75	187.69	268.10	81.41
45........	1.000	0.999	15.49	4	118.60	188.13	270.92	82.08
51........	1.000	1.000	15.47	4	122.75	188.13	272.39	82.63
44........	1.000	0.999	15.53	4	128.00	188.27	271.13	81.87
43........	1.000	0.986	15.59	4	133.50	186.24	256.84	77.79
42........	1.000	0.969	15.67	4	141.75	183.51	238.30	73.16
41........	1.000	0.791	16.48	4	188.25	153.57	....	....
9........	0.781	0.833	15.94	3	76.00	159.14	204.42	71.14
8........	0.781	0.844	15.90	4	88.50	160.91	226.14	78.03
7........	0.781	0.848	15.90	4	98.50	161.74	238.45	81.86
6........	0.781	0.850	15.90	4	103.50	162.15	243.59	83.41
5........	0.781	0.853	15.89	4	107.50	162.70	245.77	83.93
10........	0.781	0.852	15.87	4	110.75	162.42	245.76	84.17
4........	0.781	0.849	15.92	4	114.25	162.01	245.85	84.15
3........	0.781	0.842	15.98	4	119.25	161.05	240.66	82.52
2........	0.781	0.834	15.98	4	126.25	159.54	233.46	80.86
1........	0.781	0.825	16.06	4	132.75	158.18	223.16	77.56
40........	0.625	0.719	16.43	5	75.20	139.37	184.57	71.16
39........	0.625	0.725	16.37	3	89.67	140.54	205.01	78.67
38........	0.625	0.731	16.32	3	98.00	141.33	214.17	81.98
37........	0.625	0.729	16.32	4	102.75	140.93	214.19	82.22
36........	0.625	0.724	16.36	3	108.00	140.15	214.24	82.49
35........	0.625	0.723	16.33	3	109.33	139.76	213.20	82.47
34........	0.625	0.724	16.36	4	110.00	140.02	210.81	81.25
33........	0.625	0.716	16.39	3	115.00	138.58	208.79	81.16
32........	0.625	0.707	16.38	5	120.40	136.89	202.40	79.69
31........	0.625	0.692	16.44	4	127.25	134.18	192.53	77.05
30........	0.625	0.677	16.54	4	133.75	131.61	179.88	72.95
18........	0.531	0.633	16.63	4	71.50	123.52	161.07	69.22
17........	0.531	0.643	16.63	3	83.00	125.40	175.81	74.43
16........	0.531	0.642	16.63	4	93.25	125.15	184.98	78.47
15........	0.531	0.640	16.62	3	98.67	124.77	185.78	79.09
14........	0.531	0.635	16.66	3	104.00	123.90	185.32	79.28
13........	0.531	0.624	16.72	3	111.00	122.03	182.67	79.18
12........	0.531	0.613	16.75	3	115.67	120.05	175.01	76.84
11........	0.531	0.599	16.83	3	124.00	117.58	166.75	74.40
19........	0.531	0.571	16.91	4	140.25	112.33	141.46	65.75
61........	0.437	0.556	17.04	3	73.33	109.79	147.93	69.81
60........	0.437	0.561	17.07	4	86.25	110.87	162.39	75.75
59........	0.437	0.561	17.09	3	94.33	110.87	164.92	76.84
58........	0.437	0.554	17.13	3	100.67	109.67	162.47	76.35
57........	0.437	0.547	17.15	4	106.25	108.36	160.75	76.87

TABLE 85—Continued.

1	2	3	4	5	6	7	8	9
56........	0.437	0.525	17.16	4	110.25	105.97	155.69	75.59
55........	0.437	0.533	17.18	3	111.67	105.62	153.94	74.90
53........	0.437	0.525	17.24	3	116.67	104.32	149.06	73.17
54........	0.437	0.517	17.22	3	125.67	102.58	143.66	71.81
52........	0.437	0.506	17.24	4	133.00	100.57	134.15	68.31
29........	0.250	0.347	17.64	4	68.50	69.79	89.82	64.41
28........	0.250	0.346	17.66	3	77.00	69.59	93.20	66.95
27........	0.250	0.342	17.69	4	84.75	68.88	91.18	66.07
26........	0.250	0.333	17.70	4	91.50	68.06	89.22	65.38
25........	0.250	0.385	17.65	4	98.50	67.24	86.11	64.05
24........	0.250	0.332	17.66	4	102.50	66.64	82.71	62.05
22........	0.250	0.328	17.68	4	107.50	65.94	80.96	61.31
23........	0.250	0.324	17.67	3	112.00	65.23	79.08	60.57
21........	0.250	0.319	17.70	4	122.00	64.13	73.83	57.43
20........	0.250	0.308	17.77	4	142.50	62.16	47.91	38.29

TABLE 86.

Holyoke Tests of a 48-inch Left Hand Special McCormick (Jolly) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of opening of the speed-gate. In inches.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
78........	3.75	1.001	14.95	4	106.50	197.36	268.33	80.19
77........	3.75	1.001	14.89	4	111.50	196.91	269.69	81.10
76........	3.75	1.000	14.91	3	115.33	196.91	271.20	81.45
75........	3.75	1.000	14.93	4	119.25	196.91	272.41	81.70
74........	3.75	1.000	14.88	4	122.50	196.62	271.60	81.86
73........	3.75	0.999	14.90	3	126.25	196.62	271.43	81.70
72........	3.75	1.001	14.91	4	130.00	197.06	270.76	81.26
71........	3.75	1.003	14.95	4	138.50	197.65	269.85	80.53
70........	3.75	1.001	14.99	5	145.20	197.50	263.40	78.45
69........	3.75	0.828	15.75	5	202.80	167.53	....	....
68........	3.25	0.939	15.28	4	103.25	187.11	267.07	82.37
67........	3.25	0.938	15.25	3	107.00	186.68	269.59	83.50
66........	3.25	0.935	15.28	4	111.00	186.39	272.21	84.28
65........	3.25	0.935	15.29	4	114.50	186.39	273.10	84.50
64........	3.25	0.935	15.29	4	118.50	186.39	274.67	84.98
63........	3.25	0.936	15.29	4	122.25	186.68	275.15	85.00
62........	3.25	0.937	15.21	4	123.75	186.39	274.37	85.34
61........	3.25	0.936	15.18	4	126.75	185.95	272.51	85.13
60........	3.25	0.929	15.19	4	132.75	184.66	267.57	84.11
59........	3.25	0.918	15.22	4	137.00	182.64	257.73	81.75
58........	3.125	0.899	15.35	4	103.75	179.64	261.40	83.59
57........	3.125	0.899	15.35	4	109.75	179.49	265.45	84.96
56........	3.125	0.898	15.38	4	113.50	179.49	266.90	85.25
55........	3.125	0.898	15.40	4	118.50	179.64	270.69	86.28
53........	3.125	0.897	15.45	4	121.75	179.78	269.94	85.69
54........	3.125	0.896	15.43	4	123.75	179.49	270.21	86.03
52........	3.125	0.893	15.45	4	126.50	178.92	267.72	85.40
51........	3.125	0.886	15.48	3	130.00	177.78	262.03	83.95
50........	3.125	0.871	15.55	5	135.00	175.09	253.96	82.25
49........	3.00	0.871	15.53	4	106.75	175.09	261.78	84.89
48........	3.00	0.871	15.51	4	110.75	174.95	264.15	85.84
47........	3.00	0.871	15.56	3	115.00	175.09	266.56	86.27
46........	3.00	0.869	15.56	4	118.50	174.81	266.71	86.46
45........	3.00	0.867	15.55	4	121.25	174.39	264.76	86.09
44........	3.00	0.862	15.58	4	124.00	173.40	262.43	85.65
43........	3.00	0.858	15.61	3	125.00	172.84	260.35	85.09
42........	3.00	0.849	15.63	4	128.25	171.02	254.19	83.86
41........	3.00	0.839	15.64	4	132.25	169.20	248.79	82.90
40........	3.00	0.828	15.55	4	136.00	166.42	237.57	80.95
39........	2.625	0.794	15.63	4	97.75	160.09	229.86	81.00
38........	2.625	0.797	15.68	4	104.50	160.91	238.71	83.42
37........	2.625	0.797	15.70	4	109.75	160.91	243.33	84.93
36........	2.625	0.793	15.78	4	113.75	160.64	244.56	85.07
35........	2.625	0.787	15.78	4	116.50	159.41	242.64	85.05
34........	2.625	0.775	15.78	4	120.25	156.95	238.34	84.85
33........	2.625	0.766	15.85	4	124.75	155.46	234.68	83.98
32........	2.625	0.748	16.06	4	131.00	152.89	228.84	82.18

TABLE 86—Continued.

1	2	3	4	5	6	7	8	9
30........	2.25	0.709	15.45	4	92.75	141.99	193.18	77.69
31........	2.25	0.709	15.24	3	95.00	141.07	191.48	78.53
29........	2.25	0.712	15.66	4	105.25	143.69	208.60	81.74
28........	2.25	0.710	15.66	3	109.00	143.17	208.71	82.08
27........	2.25	0.697	15.77	4	115.00	141.07	208.60	82.68
26........	2.25	0.681	15.96	3	118.00	138.71	206.13	82.10
25........	2.25	0.669	16.06	4	121.50	136.76	204.08	81.93
24........	2.25	0.660	16.13	4	127.50	135.21	201.21	81.39
23........	2.25	0.653	16.17	5	134.60	133.79	194.43	79.25
21........	1.875	0.583	16.53	3	94.33	120.91	167.95	74.10
22........	1.875	0.584	16.50	4	99.00	120.91	171.61	75.85
20........	1.875	0.581	16.55	4	102.25	120.42	171.75	75.99
19........	1.875	0.574	16.59	4	107.50	119.18	169.73	75.69
18........	1.875	0.564	16.64	4	112.25	117.21	165.92	74.84
17........	1.875	0.560	16.69	4	116.00	116.60	164.42	74.50
16........	1.875	0.548	16.70	4	125.75	114.15	160.53	74.25
14........	1.875	0.541	16.78	4	135.25	112.94	154.48	71.87
15........	1.875	0.535	16.80	4	145.75	111.73	146.89	69.00
13........	1.50	0.427	17.25	4	89.50	90.31	117.26	66.37
12........	1.50	0.421	17.27	4	93.75	89.30	116.53	66.62
11........	1.50	0.415	17.28	4	100.75	87.97	115.07	66.75
10........	1.50	0.408	17.32	3	108.00	86.64	112.47	66.09
9........	1.50	0.402	17.33	4	117.00	85.32	106.12	63.28
8........	1.50	0.400	17.34	4	124.50	84.88	100.38	60.13
7........	1.50	0.400	17.38	3	139.33	84.99	93.61	55.88
4........	0.75	0.245	17.97	4	80.50	53.03	59.49	55.05
3........	0.75	0.248	18.02	4	98.75	53.60	59.71	54.51
2........	0.75	0.247	17.96	4	112.25	53.32	52.79	48.61
1........	0.75	0.243	17.95	5	123.60	52.38	41.52	38.94
5........	0.75	0.241	17.97	4	140.00	52.01	28.22	26.62
6........	0.75	0.239	17.95	4	152.50	51.73	....	....

TABLE 87.

Holyoke Tests of a 30-inch Right Hand Camden Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of opening of the speed-gate. In inches.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
30........	4.50	1.015	15.87	3	135.67	107.34	148.73	77.14
29........	4.50	1.008	15.92	4	146.25	106.86	151.89	78.88
28........	4.50	1.004	15.95	4	157.50	106.50	154.49	80.35
27........	4.50	1.000	15.97	4	168.50	106.14	155.55	81.08
26........	4.50	0.997	15.97	4	178.50	105.79	154.49	80.79
25........	4.50	0.994	16.02	3	189.33	105.67	152.93	79.82
24........	4.50	0.992	16.04	4	200.50	105.55	150.39	78.49
23........	4.50	0.993	16.03	5	211.60	105.55	146.50	76.51
22........	4.00	0.947	16.19	3	131.00	101.21	145.12	78.25
21........	4.00	0.942	16.22	4	138.50	100.74	147.83	79.94
20........	4.00	0.937	16.25	4	149.00	100.27	150.44	81.58
19........	4.00	0.933	16.26	5	161.00	99.92	153.28	83.35
18........	4.00	0.933	16.27	4	173.00	99.92	154.72	84.09
17........	4.00	0.933	16.29	3	186.00	100.04	155.61	84.37
16........	4.00	0.934	16.28	3	200.00	100.04	155.78	84.51
15........	4.00	0.934	16.28	4	217.50	100.04	150.59	81.70
14........	4.00	0.922	16.33	7	235.14	98.99	135.68	74.15
13........	4.00	0.852	16.49	4	298.50	91.91	....	....
7........	3.50	0.875	17.35	4	119.00	96.83	140.75	74.02
6........	3.50	0.867	17.42	4	138.25	96.14	151.56	79.96
5........	3.50	0.864	17.43	3	157.67	95.79	159.20	84.25
4........	3.50	0.862	17.44	4	176.25	95.57	162.71	86.25
10........	3.50	0.863	17.36	4	190.25	95.45	164.65	87.80
12........	3.50	0.859	16.54	5	179.40	92.81	152.16	87.58
11........	3.50	0.860	16.54	5	184.40	92.92	152.14	87.47
3........	3.50	0.865	17.40	4	204.00	95.79	164.78	87.35
9........	3.50	0.862	17.36	4	208.75	95.34	162.60	86.80
8........	3.50	0.858	17.39	4	213.50	95.00	160.14	85.64
2........	3.50	0.852	17.46	4	221.00	94.54	153.01	81.90
1........	3.50	0.827	17.52	4	232.75	91.93	134.29	73.67
37........	3.00	0.767	16.10	3	145.67	81.74	126.07	84.64
36........	3.00	0.767	16.15	4	162.00	81.84	130.86	87.48
38........	3.00	0.768	16.12	3	170.00	81.84	132.41	88.68
35........	3.00	0.766	16.15	3	177.00	81.74	132.76	88.86
34........	3.00	0.758	16.18	3	187.33	80.98	129.70	87.46
33........	3.00	0.745	16.26	4	194.50	79.79	123.44	84.07
32........	3.00	0.737	16.27	4	204.25	78.93	117.85	81.08
31........	3.00	0.732	16.26	4	232.75	78.40	107.43	74.46
55........	2.60	0.675	15.56	3	132.67	70.73	99.51	79.89
54........	2.60	0.680	15.56	3	152.00	71.25	107.87	85.97
56........	2.60	0.679	15.55	4	157.50	71.14	109.05	87.10
53........	2.60	0.677	15.59	3	164.67	71.04	109.26	87.17
52........	2.60	0.668	15.64	3	171.00	70.11	106.55	85.86
51........	2.60	0.659	15.65	3	178.67	69.19	103.09	84.12
50........	2.60	0.649	15.70	3	191.67	68.27	99.53	82.04
49........	2.60	0.633	15.75	4	207.25	66.74	89.68	75.38

TABLE 87—Continued.

1	2	3	4	5	6	7	8	9
47........	2.25	0.581	15.92	3	122.00	61.56	81.65	73.61
46........	2.25	0.590	15.91	4	143.00	62.45	89.93	79.97
48........	2.25	0.594	15.87	4	150.50	62.84	92.04	81.55
45........	2.25	0.589	15.94	4	160.75	62.45	94.60	83.97
44........	2.25	0.585	15.94	3	168.33	62.05	92.27	82.42
43........	2.25	0.577	15.98	4	177.50	61.26	90.12	81.34
42........	2.25	0.567	16.03	3	186.33	60.28	87.08	79.62
41........	2.25	0.561	16.07	4	194.75	59.70	83.15	76.58
40........	2.25	0.555	16.09	4	203.00	59.12	78.47	72.89
39........	2.25	0.551	16.17	4	211.25	58.83	73.13	67.92
65........	1.75	0.462	18.02	3	117.33	52.08	74.47	70.11
64........	1.75	0.459	18.03	3	126.00	51.80	76.33	72.21
63........	1.75	0.459	18.04	5	135.60	51.80	78.24	73.97
62........	1.75	0.459	18.05	4	144.75	51.80	79.34	74.98
61........	1.75	0.459	18.08	4	154.00	51.80	79.97	75.44
67........	1.75	0.459	17.99	4	154.50	51.71	78.89	74.93
66........	1.75	0.459	18.02	3	156.33	51.71	78.47	74.41
60........	1.75	0.458	18.11	4	161.75	51.71	79.33	74.84
59........	1.75	0.456	18.14	4	169.75	51.62	78.35	73.93
57........	1.75	0.452	18.15	4	177.75	51.15	76.92	73.20
58........	1.75	0.443	18.17	4	204.00	50.14	70.62	68.49

TABLE 88.

Holyoke Tests of a 36-inch Right Hand Special McCormick (Jolly) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of opening of the speed-gate. In feet.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
10........	0.354	1.004	16.29	4	160.75	129.18	192.31	80.64
9........	0.354	1.002	16.28	4	167.00	128.80	193.03	81.23
8........	0.354	1.001	16.30	3	172.00	128.80	193.84	81.47
7........	0.354	1.002	16.22	4	175.75	128.55	192.98	81.67
6........	0.354	0.999	16.20	4	179.00	128.17	191.38	81.33
5........	0.354	1.000	16.19	5	184.20	128.17	191.62	81.48
3........	0.354	1.000	16.02	4	191.25	127.54	187.90	81.15
2........	0.354	1.002	16.01	5	201.20	127.79	186.05	80.24
1........	0.354	0.819	16.50	5	282.60	105.97	....	....
18........	0.330	0.963	16.35	4	164.50	124.15	192.04	83.48
17........	0.330	0.961	16.35	4	170.25	123.90	191.86	83.57
16........	0.330	0.961	16.37	3	176.00	123.90	193.26	84.08
15........	0.330	0.962	16.36	4	180.50	124.02	192.98	83.93
14........	0.330	0.961	16.36	4	186.00	123.90	193.49	84.23
13........	0.330	0.962	16.37	4	190.75	124.02	192.92	83.85
12........	0.330	0.961	16.34	4	201.25	123.77	191.91	83.73
11........	0.330	0.956	16.39	4	215.25	123.40	186.60	81.41
27........	0.310	0.907	16.55	4	161.75	117.58	186.96	84.78
26........	0.310	0.908	16.54	3	167.00	117.70	188.20	85.30
25........	0.310	0.907	16.54	3	172.00	117.58	188.87	85.69
24........	0.310	0.909	16.52	3	178.00	117.70	190.31	86.36
23........	0.310	0.909	16.52	4	183.25	117.70	190.63	86.51
22........	0.310	0.906	16.50	3	188.00	117.33	190.14	86.66
21........	0.310	0.906	16.49	4	192.50	117.21	189.13	86.34
20........	0.310	0.899	16.48	4	199.50	116.36	184.47	84.88
19........	0.310	0.883	16.53	3	203.00	114.40	175.98	82.11
37........	0.266	0.807	16.66	4	155.25	105.03	170.47	85.96
35........	0.266	0.807	16.68	4	161.25	105.03	172.40	86.83
36........	0.266	0.807	16.69	3	165.00	105.03	173.55	87.36
34........	0.266	0.806	16.70	3	168.00	104.91	174.76	88.02
33........	0.266	0.802	16.69	4	171.00	104.44	172.94	87.54
32........	0.266	0.798	16.71	4	173.00	103.97	169.97	86.33
31........	0.266	0.786	16.74	4	179.00	102.44	165.52	85.17
30........	0.266	0.770	16.78	4	185.50	100.57	160.81	84.08
29........	0.266	0.756	16.83	4	192.00	98.83	155.35	82.41
28........	0.266	0.741	16.88	4	198.25	96.99	148.95	80.28
45........	0.228	0.702	17.05	4	140.00	92.43	141.59	79.28
44........	0.228	0.705	16.99	4	151.00	92.66	148.35	83.15
43........	0.228	0.705	16.89	4	156.50	92.32	149.23	84.45
42........	0.228	0.694	16.90	4	161.50	90.98	147.47	84.63
41........	0.228	0.682	16.93	3	167.00	89.41	144.77	84.39
40........	0.228	0.667	16.98	4	173.75	87.64	140.58	83.36
39........	0.228	0.647	17.04	4	184.50	85.10	133.29	81.10
38........	0.228	0.636	17.06	4	196.00	83.68	124.60	77.02

TABLE 88—Continued.

1	2	3	4	5	6	7	8	9
53........	0.203	0.612	17.19	4	130.75	80.85	117.12	74.36
52........	0.203	0.617	17.20	4	143.75	81.61	124.61	78.33
51........	0.203	0.615	17.22	4	150.50	81.39	126.12	79.40
50........	0.203	0.609	17.23	4	154.50	80.52	125.00	79.51
49........	0.203	0.600	17.23	4	158.50	79.34	123.66	79.82
48........	0.203	0.583	17.28	4	168.75	77.20	121.91	80.63
47........	0.203	0.570	17.30	4	177.50	75.61	117.97	79.58
46........	0.203	0.562	17.34	7	191.57	74.56	110.71	75.56
60........	0.177	0.494	17.45	4	126.25	65.73	91.20	70.16
59........	0.177	0.497	17.46	4	139.25	66.14	96.57	73.79
58........	0.177	0.495	17.50	4	143.75	65.94	95.54	73.05
57........	0.177	0.488	17.51	4	148.00	65.13	94.09	72.80
56........	0.177	0.477	17.54	4	164.75	63.73	90.45	71.40
55........	0.177	0.461	17.59	4	187.00	61.56	86.46	70.45
54........	0.177	0.455	17.60	4	197.50	60.88	79.90	65.80

TABLE 89.

Holyoke Tests of a 30-inch Right Hand Allis-Chalmers (Runner No. 22) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of opening of the speed-gate. In inches.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
108	2.875	0.993	16.71	4	166.50	95.50	144.20	79.67
107	2.875	0.992	16.73	4	180.50	95.50	145.90	80.52
106	2.875	0.995	16.74	4	189.00	95.73	147.31	81.06
105	2.875	0.997	16.70	4	197.00	95.84	147.86	81.46
104	2.875	1.002	16.70	4	205.25	96.29	148.13	81.22
102	2.875	1.006	16.74	4	216.00	96.87	149.65	81.37
103	2.875	1.013	16.69	5	224.00	97.33	148.73	80.73
101	2.875	1.023	16.75	5	237.40	98.49	150.77	80.59
8	2.75	0.974	17.20	4	149.25	95.04	146.49	79.02
7	2.75	0.973	17.23	3	163.00	95.04	150.57	81.08
6	2.75	0.974	17.25	4	178.25	95.16	154.37	82.92
5	2.75	0.976	17.23	4	188.25	95.27	155.42	83.49
4	2.75	0.978	17.15	4	196.25	95.27	155.23	83.77
3	2.75	0.983	17.13	4	208.50	95.73	156.49	84.15
9	2.75	0.992	17.20	4	220.75	96.76	159.31	84.41
10	2.75	0.997	17.21	3	230.67	97.33	159.81	84.13
2	2.75	1.000	17.05	3	235.00	97.10	156.03	83.10
1	2.75	1.007	16.95	5	256.40	97.56	148.03	78.93
19	2.50	0.904	17.32	4	149.25	88.52	137.87	79.29
18	2.50	0.909	17.32	4	168.50	88.97	145.93	83.50
17	2.50	0.915	17.32	5	186.20	89.63	150.51	85.49
16	2.50	0.920	17.32	4	196.50	90.08	153.16	86.56
25	2.50	0.922	17.26	4	202.25	90.08	154.14	87.41
15	2.50	0.926	17.30	3	207.00	90.64	155.37	87.37
26	2.50	0.926	17.24	5	209.60	90.42	154.90	87.62
14	2.50	0.929	17.28	4	216.25	90.87	156.07	87.64
23	2.50	0.929	17.25	4	216.50	90.75	156.25	88.01
20	2.50	0.935	17.27	4	227.00	91.43	157.27	87.83
22	2.50	0.936	17.24	5	227.00	91.43	157.27	87.98
24	2.50	0.935	17.24	3	233.33	91.31	154.92	86.78
13	2.50	0.936	17.27	3	234.00	91.54	155.37	86.66
12	2.50	0.928	17.28	4	243.00	90.75	147.31	82.83
21	2.50	0.927	17.25	4	244.50	90.53	145.40	82.10
11	2.50	0.913	17.31	4	255.75	89.41	132.89	75.71
44	2.25	0.842	17.65	4	157.00	83.24	135.97	81.60
43	2.25	0.854	17.68	4	180.75	84.44	146.10	86.29
42	2.25	0.857	17.72	4	190.50	84.88	148.48	87.05
35	2.25	0.858	17.45	4	192.25	84.34	146.52	87.78
34	2.25	0.861	17.41	4	195.00	84.55	146.36	87.67
33	2.25	0.863	17.40	5	199.40	84.66	147.36	88.21
41	2.25	0.864	17.68	4	204.75	85.43	152.50	89.03
36	2.25	0.866	17.45	5	205.40	85.10	149.42	88.72
40	2.25	0.866	17.44	4	207.00	85.10	149.99	89.11
32	2.25	0.868	17.43	4	210.00	85.21	151.56	89.98
37	2.25	0.868	17.46	4	211.25	85.32	151.24	89.52
38	2.25	0.868	17.44	4	212.75	85.32	151.08	89.53
39	2.25	0.868	17.44	4	215.75	85.32	150.72	89.32
31	2.25	0.864	17.38	3	221.00	84.77	146.74	87.82
30	2.25	0.852	17.35	4	229.75	83.46	139.28	84.81
29	2.25	0.847	17.35	4	240.50	83.02	131.91	80.75
28	2.25	0.851	17.37	4	264.00	83.46	121.94	74.17
27	2.25	0.809	17.47	3	342.33	79.55	....	....

TABLE 89—Continued.

1	2	3	4	5	6	7	8	9
57.......	2.125	0.811	17.86	4	152.00	80.63	131.64	80.60
56.......	2.125	0.815	17.79	3	163.67	80.85	137.02	84.00
55.......	2.125	0.820	17.78	4	180.25	81.39	142.57	86.87
53.......	2.125	0.825	17.64	3	190.00	81.50	144.80	88.81
49.......	2.125	0.828	17.60	4	213.00	81.72	145.11	88.96
46.......	2.125	0.813	17.71	4	240.00	80.52	131.64	81.40
45.......	2.125	0.815	17.74	4	263.00	80.74	121.47	74.78
117.......	2.125	0.820	17.44	4	167.75	80.52	135.59	85.14
116.......	2.125	0.825	17.44	4	180.25	81.07	140.49	87.62
115.......	2.125	0.830	17.45	4	191.25	81.61	143.54	88.88
113.......	2.125	0.835	17.54	5	202.00	82.26	148.12	90.52
114.......	2.125	0.834	17.48	5	203.00	82.04	147.09	90.44
112.......	2.125	0.836	17.65	5	208.60	82.58	149.34	90.35
111.......	2.125	0.834	17.67	4	212.50	82.47	148.45	89.83
110.......	2.125	0.832	17.62	4	215.00	82.15	146.48	89.23
109.......	2.125	0.830	17.59	4	217.00	81.93	144.08	88.15
118.......	2.125	0.821	17.43	4	224.25	80.63	135.95	85.29
119.......	2.125	0.817	17.43	4	237.25	80.20	130.13	82.08
67.......	2.00	0.781	17.28	3	151.67	76.35	122.59	81.93
66.......	2.00	0.785	17.19	3	162.00	76.56	126.27	84.60
65.......	2.00	0.789	17.20	3	174.33	76.99	130.85	87.13
63.......	2.00	0.795	17.24	3	185.33	77.63	134.82	88.83
62.......	2.00	0.797	17.30	3	193.00	77.95	137.07	89.62
64.......	2.00	0.797	17.19	4	194.25	77.74	136.26	89.91
61.......	2.00	0.798	17.34	3	199.25	78.16	138.05	89.81
60.......	2.00	0.789	17.39	3	209.33	77.42	132.94	87.07
68.......	2.00	0.782	17.51	4	223.50	76.99	129.04	84.40
59.......	2.00	0.780	17.47	4	230.00	76.67	126.16	83.05
58.......	2.00	0.781	17.53	4	261.75	76.88	113.34	74.16
78.......	1.75	0.704	17.59	4	145.75	69.49	111.08	80.13
77.......	1.75	0.708	17.63	3	156.00	69.89	115.28	82.50
76.......	1.75	0.712	17.61	4	170.25	70.31	120.90	86.10
79.......	1.75	0.717	17.55	4	177.50	70.62	122.98	87.49
75.......	1.75	0.716	17.58	4	182.50	70.62	124.33	88.31
74.......	1.75	0.713	17.57	4	188.75	70.31	123.14	87.90
73.......	1.75	0.709	17.57	3	195.00	69.89	121.59	87.31
72.......	1.75	0.706	17.57	4	201.25	69.59	119.68	86.31
71.......	1.75	0.703	17.57	3	213.00	69.28	116.83	84.63
70.......	1.75	0.699	17.59	4	229.00	68.98	112.38	81.67
69.......	1.75	0.695	17.57	4	255.00	68.57	103.06	75.43
89.......	1.50	0.621	17.76	4	147.75	61.56	98.10	79.12
90.......	1.50	0.627	17.74	4	163.25	62.16	103.68	82.90
87.......	1.50	0.628	17.74	4	174.75	62.25	105.94	84.59
86.......	1.50	0.623	17.74	4	179.50	61.76	104.67	84.24
85.......	1.50	0.622	17.75	4	184.25	61.66	104.25	83.99
84.......	1.50	0.619	17.87	4	194.00	61.56	104.17	83.49
83.......	1.50	0.616	17.75	4	201.25	61.07	102.25	83.17
82.......	1.50	0.610	17.78	4	214.50	60.49	99.07	81.23
88.......	1.50	0.606	17.79	4	233.50	60.10	94.37	77.83
81.......	1.50	0.606	17.80	4	244.50	60.10	91.76	75.63
80.......	1.50	0.599	17.81	4	267.75	59.52	77.29	64.29
97.......	1.25	0.524	17.80	4	148.50	52.01	81.45	77.58
96.......	1.25	0.528	17.79	4	161.50	52.38	83.92	79.41
95.......	1.25	0.528	17.81	6	172.71	52.47	84.76	79.97
94.......	1.25	0.524	17.83	4	181.50	52.01	83.83	79.71
93.......	1.25	0.520	17.85	4	190.75	51.64	82.60	79.01
92.......	1.25	0.507	17.87	3	210.33	50.44	78.93	77.22
98.......	1.25	0.507	17.83	4	213.75	50.35	77.75	76.36
99.......	1.25	0.504	17.84	4	228.25	50.07	72.48	71.55
91.......	1.25	0.502	17.90	5	229.00	49.98	72.72	71.67
100.......	1.25	0.507	17.84	4	249.00	50.35	64.69	62.51

TABLE 90.

Holyoke Tests of a 35-inch (Pitch Diam. 30-inch) Right Hand Samson Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
8........	1.000	0.994	17.30	3	165.67	116.57	182.75	79.91
7........	1.000	0.994	17.30	4	171.75	116.57	184.48	80.66
6........	1.000	0.995	17.28	4	175.50	116.70	185.45	81.09
5........	1.000	0.995	17.30	3	181.00	116.70	187.05	81.70
4........	1.000	0.996	17.30	4	184.50	116.82	187.46	81.79
3........	1.000	0.997	17.31	4	190.75	117.06	188.27	81.93
9........	1.000	1.000	17.30	3	197.67	117.31	189.36	82.27
2........	1.000	1.001	17.28	4	203.50	117.43	189.04	82.15
1........	1.000	1.005	17.28	4	216.25	117.80	188.33	81.58
16........	0.933	0.939	17.41	4	161.50	110.48	178.15	81.67
15........	0.933	0.942	17.42	4	176.00	110.85	183.93	83.99
14........	0.933	0.944	17.41	4	182.00	111.09	184.92	84.29
13........	0.933	0.946	17.39	3	188.00	111.33	185.56	84.51
12........	0.933	0.949	17.39	4	199.25	111.58	188.56	85.69
11........	0.933	0.952	17.37	3	206.33	111.94	189.27	85.83
10........	0.933	0.956	17.37	3	217.00	112.42	188.98	85.34
17........	0.933	0.958	17.37	4	231.25	112.67	187.97	84.69
27........	0.824	0.859	17.57	4	166.50	101.59	172.07	85.00
26........	0.824	0.863	17.57	5	176.20	102.06	175.97	86.53
24........	0.824	0.866	17.56	3	182.00	102.41	177.52	87.04
25........	0.824	0.866	17.57	4	183.50	102.41	177.92	87.19
23........	0.824	0.867	17.57	3	186.67	102.52	178.82	87.54
21........	0.824	0.868	17.55	4	192.75	102.52	179.05	87.75
22........	0.824	0.870	17.55	3	201.67	102.76	181.49	88.73
20........	0.824	0.871	17.53	3	208.00	102.87	181.14	88.57
19........	0.824	0.865	17.55	4	217.00	102.17	176.38	86.74
18........	0.824	0.852	17.61	4	221.75	100.89	167.37	83.07
35........	0.750	0.789	17.81	4	161.00	93.87	158.91	83.81
34........	0.750	0.793	17.81	4	171.50	94.44	164.29	86.13
32........	0.750	0.798	17.77	4	181.00	94.89	168.14	87.93
33........	0.750	0.798	17.78	4	184.00	94.89	168.79	88.22
31........	0.750	0.798	17.77	4	195.00	94.89	169.82	88.81
30........	0.750	0.789	17.81	3	202.00	93.87	164.19	86.60
29........	0.750	0.782	17.81	3	204.33	93.07	160.15	85.19
28........	0.750	0.774	17.83	5	207.00	92.17	156.24	83.83
45........	0.700	0.734	17.88	4	152.50	87.49	145.21	81.85
44........	0.700	0.742	17.88	4	162.75	88.49	151.19	84.26
42........	0.700	0.748	17.91	4	175.50	89.26	157.94	87.11
43........	0.700	0.750	17.86	4	180.25	89.37	159.07	87.88
41........	0.700	0.749	17.90	4	185.25	89.37	160.26	88.33
40........	0.700	0.746	17.91	4	187.75	89.04	158.06	87.40
39........	0.700	0.741	17.94	4	191.75	88.49	155.86	86.57
38........	0.700	0.733	17.98	4	194.75	87.71	152.64	85.35
37........	0.700	0.709	17.98	4	203.00	84.84	147.32	85.16
36........	0.700	0.703	18.00	4	221.25	84.08	141.30	82.33

TABLE 90—Continued.

1	2	3	4	5	6	7	8	9
55........	0.600	0.631	18.08	4	150.00	75.67	125.41	80.83
54........	0.600	0.639	18.04	4	162.50	76.52	131.14	83.77
53........	0.600	0.640	18.04	4	169.00	76.62	131.48	83.88
52........	0.600	0.635	18.05	4	173.00	76.09	130.57	83.83
51........	0.600	0.627	18.07	4	175.00	75.15	129.04	83.79
50........	0.600	0.623	18.08	4	180.50	74.73	128.90	84.12
49........	0.600	0.621	18.10	4	186.00	74.52	128.51	84.01
48........	0.600	0.620	18.10	4	187.50	74.41	128.46	84.10
47........	0.600	0.616	18.11	4	199.50	73.89	127.41	83.96
46........	0.600	0.614	18.12	4	214.00	73.68	124.25	82.06
65........	0.500	0.526	18.26	3	148.33	63.39	103.34	78.72
64........	0.500	0.523	18.27	3	158.00	63.09	105.49	80.70
66........	0.500	0.522	18.27	4	161.50	62.99	105.95	81.18
63........	0.500	0.521	18.28	4	166.25	62.89	106.18	81.44
62........	0.500	0.521	18.30	4	174.75	62.89	106.53	81.62
60........	0.500	0.520	18.29	3	183.67	62.70	105.57	81.17
61........	0.500	0.520	18.30	4	186.75	62.79	105.17	80.71
59........	0.500	0.517	18.30	4	192.50	62.40	105.06	81.12
58........	0.500	0.506	18.33	3	198.00	61.12	100.01	78.73
56........	0.500	0.498	18.35	3	207.00	60.14	96.15	76.82
57........	0.500	0.493	18.35	3	222.00	59.56	90.22	72.79

TABLE 91.

Holyoke Tests of a 33-inch Left Hand Smith (Type O) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
44........	1.000	0.989	14.02	3	140.00	122.03	157.18	81.15
43........	1.000	0.990	14.03	3	150.67	122.28	160.48	82.63
42........	1.000	0.991	14.02	4	161.25	122.28	162.46	83.71
38........	1.000	0.996	14.07	4	174.00	123.15	165.29	84.27
37........	1.000	1.000	14.08	7	182.00	123.65	167.65	85.06
39........	1.000	1.004	14.03	3	187.00	123.90	166.88	84.80
40........	1.000	1.004	14.02	3	192.33	123.90	166.10	84.46
41........	1.000	1.007	13.96	4	197.00	124.02	164.46	83.91
45........	1.000	0.801	14.37	4	254.75	100.11	....	....
91........	1.000	0.996	16.13	5	196.80	131.86	203.95	84.70
92........	1.000	1.000	16.11	4	203.50	132.25	205.04	85.01
93........	1.000	1.001	16.11	4	208.75	132.37	204.31	84.63
94........	1.000	1.003	16.08	3	214.33	132.63	203.60	84.33
33........	0.952	0.948	14.48	4	131.25	118.94	154.91	79.45
32........	0.952	0.950	14.49	4	145.50	119.18	163.35	83.56
31........	0.952	0.952	14.48	4	155.75	119.43	165.89	84.74
30........	0.952	0.956	14.47	4	168.00	119.80	169.27	86.25
29........	0.952	0.961	14.47	4	181.25	120.54	172.18	87.20
35........	0.952	0.962	14.33	4	182.00	120.05	169.75	87.16
34........	0.952	0.964	14.36	4	185.00	120.42	170.42	87.05
36........	0.952	0.964	14.33	3	186.00	120.29	169.20	86.70
38........	0.952	0.968	14.46	4	194.50	121.28	173.57	87.43
27........	0.952	0.958	14.52	4	204.75	120.29	165.03	83.46
26........	0.952	0.932	14.60	4	212.50	117.33	146.81	75.71
11........	0.857	0.855	15.16	3	112.67	109.67	132.98	70.65
10........	0.857	0.868	15.14	4	129.50	111.36	149.12	78.13
9........	0.857	0.883	15.08	5	151.80	113.06	166.05	86.03
8........	0.857	0.888	15.08	5	164.20	113.67	170.16	87.69
6........	0.857	0.890	15.11	5	172.20	114.03	173.50	88.95
5........	0.857	0.892	15.11	4	177.75	114.28	173.97	89.00
13........	0.857	0.892	14.98	4	175.75	113.79	172.01	89.14
12........	0.857	0.894	14.99	4	180.75	114.15	173.69	89.71
7........	0.857	0.894	15.09	4	184.00	114.52	174.79	89.35
4........	0.857	0.892	15.13	4	188.00	114.40	173.18	88.38
3........	0.857	0.882	15.16	4	193.50	113.18	167.10	86.03
2........	0.857	0.862	15.22	4	201.50	110.87	156.61	81.98
1........	0.857	0.843	15.28	6	209.50	108.59	144.74	77.05
87........	0.806	0.819	16.61	4	133.50	110.03	162.18	78.38
86........	0.806	0.831	16.59	4	148.50	111.60	175.27	83.62
85........	0.806	0.844	16.57	4	166.00	113.18	186.36	87.78
82........	0.806	0.845	16.58	6	172.50	113.42	188.70	88.64
83........	0.806	0.847	16.58	4	177.75	113.67	191.37	89.70
88........	0.806	0.847	16.56	5	178.20	113.67	190.83	89.55
81........	0.806	0.847	16.56	4	180.25	113.67	191.99	90.09
84........	0.806	0.846	16.57	5	180.00	113.55	190.68	89.52
89........	0.806	0.848	16.52	5	181.00	113.67	191.74	90.20
90........	0.806	0.848	16.51	5	181.60	113.55	191.33	90.15
80........	0.806	0.842	16.60	5	186.40	113.06	187.80	88.39
79........	0.806	0.832	16.65	4	192.50	111.85	182.87	86.74
78........	0.806	0.821	16.66	4	198.00	110.51	176.69	84.77
77........	0.806	0.805	16.68	4	207.50	108.36	167.25	81.74
76........	0.806	0.781	16.70	4	216.00	105.15	149.23	75.07

TABLE 91—Continued.

1	2	3	4	5	6	7	8	9
23........	0.762	0.782	15.08	3	127.33	100.11	133.42	78.07
22........	0.762	0.797	15.06	4	142.25	101.89	144.96	83.45
21........	0.762	0.802	15.05	3	149.00	102.56	150.12	85.91
20........	0.762	0.807	15.06	4	155.50	103.26	153.99	87.47
24........	0.762	0.811	14.95	4	162.50	103.38	156.24	89.30
25........	0.762	0.811	14.95	4	165.25	103.38	156.98	89.72
19........	0.762	0.804	15.13	4	166.50	103.14	158.17	89.53
18........	0.762	0.796	15.18	4	171.75	102.21	155.25	88.38
17........	0.762	0.787	15.20	4	175.75	101.16	151.78	87.19
16........	0.762	0.777	15.23	4	182.50	99.99	147.10	85.33
15........	0.762	0.762	15.25	4	193.25	98.02	139.08	82.18
14........	0.762	0.737	15.31	5	201.80	95.04	127.80	77.59
60........	0.625	0.677	14.22	3	129.33	84.12	107.97	79.73
67........	0.625	0.680	14.05	4	135.00	84.01	110.37	82.60
59........	0.625	0.691	14.20	4	145.00	85.87	116.87	84.67
61........	0.625	0.686	14.17	4	149.25	85.10	116.00	84.98
62........	0.625	0.671	14.18	4	157.75	83.24	113.53	84.96
63........	0.625	0.652	14.19	3	166.00	80.96	109.91	84.51
64........	0.625	0.636	14.21	3	180.00	79.02	103.63	81.52
65........	0.625	0.622	14.22	3	193.67	77.31	89.20	71.67
66........	0.625	0.605	14.24	4	205.75	75.19	71.07	58.64
74........	0.499	0.549	14.18	3	128.67	68.16	85.19	77.86
73........	0.499	0.549	14.24	4	136.75	68.26	86.60	78.70
72........	0.499	0.537	14.27	3	150.00	66.84	86.36	79.98
75........	0.499	0.543	14.18	4	159.50	66.34	85.40	80.19
71........	0.499	0.529	14.31	3	170.67	65.94	83.52	78.19
70........	0.499	0.518	14.35	3	184.00	64.73	74.15	70.52
69........	0.499	0.506	14.39	4	194.25	63.24	61.51	59.71
68........	0.499	0.495	14.45	4	204.25	62.06	47.04	46.33

TABLE 92.

Holyoke Tests of a 28-inch Right Hand Wellman-Seaver-Morgan Co. Turbine. Wheel supported by Ball-bearing step. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
95.......	1.077	0.971	17.11	4	153.00	97.00	125.20	66.52
94.......	1.077	1.017	16.97	3	199.67	101.16	147.66	75.84
93.......	1.077	1.036	16.94	3	224.33	102.98	156.38	79.04
92.......	1.077	1.053	16.89	3	239.33	104.50	159.58	79.72
91.......	1.077	1.061	16.87	3	247.33	105.22	161.17	80.06
89.......	1.077	1.068	16.81	3	253.67	105.70	161.45	80.12
90.......	1.077	1.072	16.80	3	259.00	106.08	161.71	80.01
88.......	1.077	1.079	16.82	4	267.50	106.82	162.15	79.58
87.......	1.077	1.026	17.05	3	294.67	102.27	125.03	63.23
86.......	1.000	0.913	17.28	3	147.00	91.65	120.29	66.98
85.......	1.000	0.957	17.16	2	190.50	95.70	144.34	77.50
84.......	1.000	0.972	17.14	3	211.67	97.10	152.69	80.89
83.......	1.000	0.981	17.13	3	225.00	98.03	156.85	82.36
82.......	1.000	0.990	17.11	3	232.67	98.90	157.96	82.31
80.......	1.000	0.996	17.09	4	240.25	99.43	160.19	83.13
81.......	1.000	1.003	17.07	3	247.33	100.07	161.17	83.19
79.......	1.000	1.004	17.07	4	252.25	100.14	160.55	82.82
78.......	1.000	1.001	17.13	4	259.00	100.00	157.00	80.82
77.......	1.000	0.983	17.22	3	268.33	98.43	146.39	76.15
76.......	1.000	0.911	17.47	4	293.50	91.96	106.75	58.59
106.......	0.923	0.868	17.32	4	143.25	87.24	115.49	67.39
105.......	0.923	0.899	17.24	5	176.00	90.12	135.49	76.90
104.......	0.923	0.920	17.15	4	201.00	91.96	146.21	81.74
101.......	0.923	0.931	16.93	5	213.20	92.52	148.62	83.66
100.......	0.923	0.936	16.93	6	220.67	92.96	150.48	84.31
99.......	0.923	0.942	16.91	4	227.25	93.51	151.52	84.50
102.......	0.923	0.945	16.93	4	232.00	93.82	151.88	84.31
103.......	0.923	0.945	17.04	4	235.50	94.14	152.74	83.96
98.......	0.923	0.945	16.93	4	237.75	93.82	151.32	84.00
97.......	0.923	0.924	17.02	4	254.50	92.04	138.84	78.15
96.......	0.923	0.823	17.25	4	288.25	82.47	87.36	54.15
42.......	0.923	0.870	17.17	3	146.33	87.02	116.20	68.57
41.......	0.923	0.895	17.10	4	170.00	89.37	130.87	75.51
40.......	0.923	0.921	17.04	4	202.75	91.80	146.25	82.44
39.......	0.923	0.928	16.99	3	208.67	92.35	147.99	83.17
35.......	0.923	0.932	17.03	4	216.25	92.81	150.75	84.10
38.......	0.923	0.937	16.97	4	220.00	93.20	151.36	84.38
36.......	0.923	0.939	17.01	4	223.75	93.44	152.58	84.65
34.......	0.923	0.940	17.02	4	226.25	93.66	150.86	83.45
37.......	0.923	0.944	16.97	3	238.33	93.90	151.69	83.94
33.......	0.923	0.921	17.13	4	256.25	92.05	139.80	78.18
32.......	0.923	0.823	17.27	3	288.00	82.54	87.29	53.99
31.......	0.923	0.730	17.50	4	334.75	73.75	....	....
74.......	0.846	0.824	17.46	3	158.67	83.15	120.23	73.02
75.......	0.846	0.836	17.46	3	175.67	84.35	129.91	77.78
72.......	0.846	0.861	17.34	5	202.20	86.50	143.40	84.30
70.......	0.846	0.865	17.33	4	209.00	86.95	145.69	85.25
71.......	0.846	0.866	17.33	3	215.00	87.24	147.27	85.89
73.......	0.846	0.870	17.34	4	219.25	87.47	148.19	86.15
69.......	0.846	0.869	17.32	4	221.25	87.32	147.53	86.01
68.......	0.846	0.866	17.33	4	227.75	87.02	144.96	84.76
67.......	0.846	0.858	17.36	4	231.75	86.25	140.48	82.73
66.......	0.846	0.845	17.39	4	243.75	85.11	132.98	79.22
65.......	0.846	0.828	17.44	4	256.50	83.44	124.39	75.37
64.......	0.846	0.754	17.59	3	282.00	76.31	85.47	56.15

TABLE 92—Continued.

1	2	3	4	5	6	7	8	9
30.......	0.769	0.750	17.38	3	141.00	75.52	102.56	68.90
26.......	0.769	0.766	17.35	3	166.00	77.02	115.72	76.36
27.......	0.769	0.779	17.32	3	183.00	78.24	124.24	80.84
25.......	0.769	0.789	17.31	3	194.00	79.25	129.36	83.15
29.......	0.769	0.793	17.25	4	200.75	79.48	131.42	84.52
28.......	0.769	0.792	17.26	4	206.00	79.40	131.11	84.36
24.......	0.769	0.773	17.38	4	226.25	77.82	123.43	80.47
23.......	0.769	0.735	17.49	3	251.33	74.17	106.64	72.49
22.......	0.769	0.690	17.56	3	269.67	69.82	81.73	58.78
21.......	0.769	0.623	17.68	4	323.75	63.27	....	....
20.......	0.615	0.617	17.91	4	139.50	63.07	88.79	69.31
16.......	0.615	0.627	17.79	3	158.33	63.81	95.97	74.55
17.......	0.615	0.634	17.80	3	171.33	64.55	101.26	77.71
15.......	0.615	0.638	17.73	4	179.50	64.82	103.37	79.31
18.......	0.615	0.636	17.77	4	183.00	64.74	103.16	79.07
19.......	0.615	0.634	17.80	2	188.00	64.61	102.56	78.64
14.......	0.615	0.627	17.72	4	194.50	63.74	100.21	78.24
13.......	0.615	0.596	17.77	3	218.67	60.60	92.78	75.97
12.......	0.615	0.563	17.79	4	243.00	57.29	73.65	63.72
11.......	0.615	0.519	17.92	4	312.00	53.00	....	....
10.......	0.462	0.452	17.34	3	117.33	45.40	56.90	63.73
5.......	0.462	0.458	17.02	3	136.00	45.65	61.83	70.17
7.......	0.462	0.461	17.08	4	146.75	46.04	64.94	72.82
6.......	0.462	0.462	17.05	4	152.00	46.04	66.34	74.52
4.......	0.462	0.462	16.92	4	155.25	45.90	65.88	74.79
9.......	0.462	0.459	17.16	3	162.00	45.94	66.78	74.69
8.......	0.462	0.457	17.15	3	166.67	45.69	65.67	73.90
3.......	0.462	0.451	16.98	4	172.25	44.83	62.65	72.57
2.......	0.462	0.432	17.05	4	217.50	43.04	52.74	63.37
1.......	0.462	0.404	17.18	5	282.40	40.40	....	....

TABLE 93.

Sam son (Type F) Turbine. Swing-gate. Conical Draft-tube.
Holyoke Tests of a 35-inch (Pitch Diam. 30-inch) Right Hand Improved Sam-

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
53........	1.000	0.991	17.23	3	173.00	122.65	205.23	85.71
54........	1.000	0.993	17.21	3	181.33	122.78	206.72	86.34
52........	1.000	0.994	17.16	4	190.00	122.78	208.91	87.51
51........	1.000	0.997	17.04	4	199.50	122.65	207.81	87.76
50........	1.000	0.999	17.05	3	206.00	122.90	208.62	87.87
49........	1.000	1.000	17.02	5	212.60	123.03	209.15	88.16
48........	1.000	1.002	17.03	4	218.50	123.27	208.63	87.71
47........	1.000	1.004	17.04	5	225.80	123.52	209.07	87.67
46........	1.000	1.008	17.03	4	237.75	124.02	206.37	86.24
45........	0.889	0.900	17.26	4	156.75	111.48	185.95	85.30
44........	0.889	0.907	17.28	4	171.50	112.33	193.53	88.00
43........	0.889	0.911	17.28	4	185.75	112.82	198.86	90.03
40........	0.889	0.913	17.23	3	192.33	112.94	200.34	90.86
39........	0.889	0.915	17.22	4	200.00	113.18	202.54	91.72
42........	0.889	0.917	17.27	4	204.00	113.55	203.64	91.65
38........	0.889	0.917	17.20	4	206.25	113.42	202.90	91.81
41........	0.889	0.917	17.25	5	211.40	113.55	203.07	91.50
37........	0.889	0.916	17.20	4	216.50	113.18	200.46	90.88
36........	0.889	0.902	17.21	4	220.75	111.48	191.62	88.15
25........	0.833	0.840	17.43	4	150.25	104.56	172.16	83.37
23........	0.833	0.847	17.43	4	164.75	105.38	181.14	87.04
24........	0.833	0.851	17.42	4	175.75	105.85	188.15	90.06
21........	0.833	0.857	17.42	4	184.50	106.56	192.18	91.38
18........	0.833	0.850	17.46	4	205.75	105.85	190.50	90.98
17........	0.833	0.835	17.52	4	209.75	104.20	182.07	88.02
74........	0.833	0.858	17.18	3	188.67	105.97	191.07	92.63
73........	0.833	0.858	17.19	4	192.00	106.09	191.66	92.76
72........	0.833	0.857	17.19	4	195.25	105.97	192.08	93.07
75........	0.833	0.858	17.18	3	198.33	105.97	191.67	92.92
35........	0.778	0.794	17.53	4	148.50	99.06	162.42	82.55
34........	0.778	0.807	17.53	4	167.50	100.69	175.44	87.73
33........	0.778	0.808	17.53	3	179.00	100.81	181.27	90.53
31........	0.778	0.812	17.47	3	186.00	101.16	182.98	91.38
32........	0.778	0.811	17.54	4	190.00	101.27	184.72	91.78
30........	0.778	0.812	17.44	4	191.50	101.04	182.85	91.58
29........	0.778	0.798	17.46	4	196.50	99.41	176.25	89.62
28........	0.778	0.781	17.54	4	203.25	97.45	170.55	88.06
27........	0.778	0.764	17.57	4	211.75	95.50	165.42	87.01
26........	0.778	0.761	17.58	5	229.00	95.16	159.02	83.90
62........	0.667	0.683	17.67	4	158.75	85.54	146.90	85.83
63........	0.667	0.689	17.66	3	165.00	86.31	149.91	86.80
61........	0.667	0.691	17.65	3	170.67	86.53	153.08	88.47
60........	0.667	0.686	17.70	4	174.25	85.98	151.25	87.72
59........	0.667	0.679	17.73	3	178.00	85.21	149.36	87.25
58........	0.667	0.669	17.75	4	182.50	84.01	147.85	87.51
57........	0.667	0.663	17.75	3	195.67	83.24	147.20	87.93
56........	0.667	0.660	17.76	4	208.75	82.91	144.96	86.89
55........	0.667	0.646	17.84	6	230.00	81.39	133.10	80.90
71........	0.556	0.559	17.81	3	148.00	70.31	115.62	81.49
70........	0.556	0.562	17.81	3	159.33	70.72	119.86	83.99
69........	0.556	0.558	17.85	3	165.00	70.31	119.35	83.94
68........	0.556	0.556	17.90	3	172.00	70.10	119.44	84.01
67........	0.556	0.554	17.92	3	189.33	69.89	120.52	84.93
66........	0.556	0.542	17.95	4	199.75	68.47	115.59	83.01
65........	0.556	0.531	17.95	3	217.67	67.04	107.07	78.53
64........	0.556	0.521	18.00	3	235.33	65.94	95.33	70.89

TABLE 94.

Holyoke Tests of a 30-inch Right Hand Allis-Chalmers (Type No. 12) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of opening of the speed-gate. In inches.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
11........	4.34	1.004	16.40	4	148.50	137.44	197.45	77.24
10........	4.34	1.001	16.38	4	162.25	136.92	201.67	79.29
9........	4.34	0.998	16.38	3	175.00	136.53	202.34	79.78
8........	4.34	1.000	16.37	4	184.75	136.66	202.93	79.99
7........	4.34	1.000	16.36	4	189.75	136.66	202.94	80.04
6........	4.34	1.001	16.36	4	195.00	136.79	202.92	79.95
5........	4.34	1.002	16.33	4	199.75	136.79	202.09	79.77
4........	4.34	1.003	16.32	4	205.00	136.92	201.47	79.50
3........	4.34	1.003	16.25	4	209.00	136.66	199.36	79.16
2........	4.34	1.011	16.22	4	225.75	137.57	195.76	77.36
13........	4.34	1.019	16.37	4	242.00	139.27	195.87	75.75
12........	4.34	1.029	16.34	4	258.50	140.57	186.80	71.71
1........	4.34	0.990	16.29	6	331.17	134.98	....	......
23........	4.125	0.955	16.59	3	152.67	131.38	194.17	78.55
22........	4.125	0.956	16.64	4	164.50	131.76	199.71	80.32
21........	4.125	0.957	16.63	4	175.50	131.89	202.92	81.58
20........	4.125	0.960	16.59	4	187.50	132.15	205.95	82.83
19........	4.125	0.963	16.51	3	198.00	132.27	206.04	83.19
18........	4.125	0.965	16.46	5	208.20	132.27	204.62	82.87
17........	4.125	0.968	16.35	4	218.00	132.27	201.65	82.22
16........	4.125	0.976	16.26	3	229.00	132.92	198.58	81.02
15........	4.125	0.981	16.25	3	242.00	133.69	195.87	79.50
14........	4.125	0.989	16.26	5	255.00	134.72	191.64	77.14
34........	3.75	0.909	16.75	4	165.75	125.68	196.44	82.28
33........	3.75	0.916	16.71	4	186.50	126.56	204.86	85.41
32........	3.75	0.919	16.69	5	191.60	126.81	204.92	85.37
31........	3.75	0.921	16.68	4	198.25	127.06	206.30	85.83
30........	3.75	0.921	16.66	3	205.00	127.06	207.40	86.39
29........	3.75	0.924	16.65	4	209.75	127.44	206.14	85.66
28........	3.75	0.926	16.63	4	216.50	127.57	206.52	85.84
27........	3.75	0.931	16.61	5	227.80	128.20	204.13	84.56
26........	3.75	0.937	16.59	3	241.00	128.96	202.02	83.26
25........	3.75	0.928	16.64	4	255.25	127.94	184.46	76.40
24........	3.75	0.909	16.72	4	344.24	125.55	....	......
75........	3.50	0.846	16.71	4	146.25	116.87	173.33	78.26
74........	3.50	0.854	16.68	4	164.25	117.85	185.16	83.06
73........	3.50	0.859	16.68	4	179.50	118.59	191.98	85.58
72........	3.50	0.864	16.66	4	191.50	119.21	195.95	87.00
71........	3.50	0.867	16.67	3	200.00	119.58	196.56	86.94
70........	3.50	0.869	16.67	4	207.00	119.82	197.46	87.17
69........	3.50	0.871	16.66	4	213.50	120.07	197.48	87.05
68........	3.50	0.873	16.65	4	219.75	120.32	196.91	86.67
67........	3.50	0.873	16.68	4	231.25	120.44	193.85	85.08
66........	3.50	0.867	16.73	3	238.00	119.82	185.75	81.71
65........	3.50	0.854	16.80	4	247.00	118.34	171.35	76.00
64........	3.50	0.844	16.88	4	344.00	117.23	....	....
62........	3.19	0.799	16.78	4	139.25	110.65	159.39	75.70
61........	3.19	0.807	16.74	4	159.00	111.50	174.65	82.51
60........	3.19	0.816	16.72	4	180.25	112.72	187.56	87.76
63........	3.19	0.820	16.73	4	187.75	113.32	189.95	88.34
59........	3.19	0.820	16.71	5	193.60	113.20	190.27	88.69
58........	3.19	0.820	16.73	3	199.00	113.32	189.82	88.29
57........	3.19	0.821	16.74	4	205.50	113.67	190.08	88.16
56........	3.19	0.822	16.75	5	211.60	113.69	189.61	87.80
55........	3.19	0.821	16.80	5	221.00	113.69	185.26	85.53
54........	3.19	0.810	16.88	4	228.25	112.47	178.14	82.74
53........	3.19	0.800	16.93	4	245.50	111.26	163.22	76.40

TABLE 94—Continued.

1	2	3	4	5	6	7	8	9
42........	2.875	0.745	16.67	4	156.50	102.81	158.33	81.46
43........	2.875	0.750	16.66	3	169.00	103.39	166.09	85.03
44........	2.875	0.753	16.65	4	175.00	103.87	169.46	86.40
41........	2.875	0.753	16.65	4	181.00	103.87	172.65	88.03
45........	2.875	0.756	16.63	4	187.00	104.22	172.97	88.00
40........	2.875	0.758	16.68	4	196.50	104.57	176.08	89.01
39........	2.875	0.752	16.79	4	206.00	104.10	172.68	87.12
38........	2.875	0.743	16.94	4	214.25	103.39	167.21	84.18
37........	2.875	0.739	17.20	4	228.75	103.51	165.30	81.87
36........	2.875	0.739	17.20	4	260.25	103.51	150.45	74.52
35........	2.875	0.717	17.27	3	332.67	100.71	....	....
97........	2.50	0.627	17.21	4	141.00	87.87	130.42	76.05
96........	2.50	0.645	17.20	4	162.50	90.32	145.61	82.65
94........	2.50	0.663	17.20	4	176.50	92.90	153.06	84.46
98........	2.50	0.663	17.13	4	185.00	92.78	155.08	86.04
93........	2.50	0.658	17.23	4	190.00	92.33	153.78	85.24
92........	2.50	0.650	17.27	4	199.50	91.33	149.93	83.82
91........	2.50	0.644	17.30	4	209.75	90.54	145.51	81.91
90........	2.50	0.640	17.33	4	220.25	90.09	140.06	79.10
89........	2.50	0.636	17.37	4	230.50	89.53	133.26	75.56
88........	2.50	0.630	17.39	4	238.75	88.76	124.22	70.96
87........	2.50	0.625	17.38	5	254.40	88.09	110.30	63.53
86........	2.50	0.617	17.34	4	272.75	86.76	78.84	46.21
84........	1.875	0.492	17.45	4	142.00	69.39	98.51	71.74
85........	1.875	0.494	17.40	4	147.00	69.59	99.43	72.41
83........	1.875	0.500	17.45	4	159.50	70.62	104.20	74.56
82........	1.875	0.504	17.38	4	166.25	70.93	103.80	74.25
81........	1.875	0.503	17.42	4	173.50	70.93	103.31	73.73
80........	1.875	0.502	17.46	4	192.00	70.93	105.45	75.08
79........	1.875	0.498	17.53	5	206.60	70.52	101.52	72.40
77........	1.875	0.493	17.79	4	226.00	70.21	91.46	64.56
78........	1.875	0.484	17.83	4	247.00	69.08	71.40	51.11
76........	1.875	0.471	17.85	3	286.33	67.25	....	....

TABLE 95.

Holyoke Tests of a 25-inch Right Hand Wellman-Seaver-Morgan Co. (Runner No. 37) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of opening of the speed-gate. In inches.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
41........	3.58	0.957	16.45	4	238.00	96.52	143.72	79.95
42........	3.58	0.974	16.42	3	256.33	98.16	147.05	80.58
40........	3.58	0.998	16.35	4	278.00	100.36	151.09	81.33
43........	3.58	1.029	16.31	3	309.67	103.28	149.60	78.44
44........	3.58	1.031	16.25	3	325.33	103.28	137.52	72.37
10........	3.30	0.858	17.25	3	170.00	88.62	128.32	74.14
6........	3.30	0.864	17.20	3	183.33	89.07	132.85	76.59
9........	3.30	0.875	17.21	3	200.33	90.21	139.12	79.15
5........	3.30	0.888	17.19	3	218.67	91.51	145.25	81.56
8........	3.30	0.904	17.14	3	236.33	93.05	149.85	82.98
4........	3.30	0.923	17.12	3	257.33	94.97	155.40	84.41
7........	3.30	0.945	17.03	3	278.00	96.98	159.48	85.29
3........	3.30	0.954	17.05	3	291.67	97.92	158.52	83.86
2........	3.30	0.954	17.03	4	303.50	97.84	148.07	78.49
1........	3.30	0.952	17.02	5	322.80	97.61	136.45	72.54
17........	2.89	0.779	17.32	3	161.67	80.56	117.15	74.16
16........	2.89	0.789	17.31	3	179.67	81.59	124.77	78.03
18........	2.89	0.803	17.24	3	198.67	82.85	131.97	81.59
15........	2.89	0.817	17.24	3	221.00	84.35	140.13	85.11
19........	2.89	0.837	17.17	3	243.67	86.21	147.15	87.80
21........	2.89	0.845	17.10	4	253.50	86.89	149.26	88.72
14........	2.89	0.846	17.17	3	259.67	87.19	148.97	87.89
20........	2.89	0.847	17.13	4	267.50	87.11	145.38	86.05
13........	2.89	0.840	17.21	4	275.50	86.66	141.41	83.74
22........	2.89	0.841	17.11	3	284.00	86.51	137.20	81.87
12........	2.89	0.843	17.21	4	296.00	86.89	134.06	79.18
11........	2.89	0.849	17.22	6	318.33	87.56	124.95	73.19
38........	2.73	0.794	16.77	4	220.75	80.85	133.30	86.84
39........	2.73	0.805	16.74	3	232.33	81.89	136.79	88.13
37........	2.73	0.810	16.72	3	241.67	82.33	138.64	88.95
36........	2.73	0.811	16.75	3	252.00	82.48	136.96	87.56
35........	2.73	0.808	16.81	3	259.67	82.33	133.29	85.06
34........	2.73	0.807	16.81	3	269.67	82.26	130.28	83.21
33........	2.47	0.695	17.30	3	159.33	71.82	105.84	75.24
32........	2.47	0.704	17.29	3	180.00	72.82	114.13	80.06
31........	2.47	0.716	17.25	3	201.67	73.96	121.78	84.31
30........	2.47	0.733	17.22	4	224.75	75.62	128.94	87.45
29........	2.47	0.732	17.25	3	234.33	75.62	127.36	86.23
28........	2.47	0.729	17.28	3	244.67	75.33	125.59	85.21
27........	2.47	0.730	17.30	5	256.60	75.47	123.96	83.86
26........	2.47	0.732	17.29	3	269.00	75.62	121.84	82.30
25........	2.47	0.733	17.34	3	281.00	75.83	118.78	79.79
23........	2.47	0.730	17.35	5	302.20	75.54	109.49	73.79
24........	2.47	0.732	17.33	3	327.33	75.76	98.83	66.49
50........	2.06	0.595	16.98	3	169.67	60.99	92.21	78.65
51........	2.06	0.607	16.95	3	194.00	62.14	99.58	83.50
49........	2.06	0.607	16.96	3	207.00	62.14	100.00	83.81
48........	2.06	0.603	17.00	3	230.67	61.80	97.51	81.97
47........	2.06	0.607	17.00	3	260.33	62.21	94.32	78.77
46........	2.06	0.612	17.03	3	292.00	62.82	88.17	72.79
45........	2.06	0.613	17.04	4	321.00	62.89	77.54	63.90

TABLE 95—Continued.

1	2	3	4	5	6	7	8	9
59........	1.65	0.477	17.11	3	145.00	49.02	69.17	72.84
58........	1.65	0.491	17.09	3	178.67	50.47	77.68	79.55
57........	1.65	0.490	17.12	3	195.00	50.40	76.54	78.35
56........	1.65	0.487	17.16	3	214.00	50.15	74.95	76.93
60........	1.65	0.483	17.11	3	232.67	49.65	71.66	74.50
55........	1.65	0.482	17.17	3	233.33	49.65	71.86	74.45
54........	1.65	0.485	17.16	3	265.33	49.90	70.50	72.72
53........	1.65	0.487	17.17	3	288.00	50.15	64.35	66.00
52........	1.65	0.486	17.21	6	307.67	50.15	55.74	57.04
66........	1.24	0.362	17.24	3	153.33	37.40	53.70	73.56
65........	1.24	0.361	17.26	3	173.00	37.29	53.28	73.11
64........	1.24	0.361	17.28	3	192.67	37.29	51.19	70.17
62........	1.24	0.358	17.31	3	219.00	37.00	48.93	67.48
61........	1.24	0.351	17.39	4	239.75	36.43	43.43	60.55
63........	1.24	0.347	17.33	3	269.00	35.91	32.49	46.11
	Jacket Removed for Remaining Experiments.							
67........	3.58	0.983	14.44	5	405.00	92.82	....	....
68........	3.30	0.958	14.46	3	418.67	90.59	....	....
69........	2.89	0.862	14.64	4	414.75	82.04	....	....
70........	2.47	0.721	14.82	4	397.25	69.01	....	....
71........	2.06	0.563	15.11	4	361.00	54.38	....	....
72........	1.65	0.456	15.22	4	338.75	44.26	....	....
73........	1.24	0.346	15.42	4	306.00	33.78	....	....

TABLE 96.

Holyoke Tests of a 25-inch Right Hand I. P. Morris (Runner "O") Turbine. Swing-gate. Conical Draft-tube.

Number of the experi-ment.	Proportional part of opening of the speed-gate. In inches.	the full dis-charge of the wheel. Per cent.	Head acting on the wheel. In feet.	Dura-tion of the experi-ment. In min.	Revolu-tions of the wheel. Per min.	Quan-tity of water dis-charged by the wheel. Cubic ft. per sec.	Power devel-oped by the wheel. H. P.	Effic-iency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
55........	4.00	0.917	17.74	3	182.33	97.79	146.49	74.62
54........	4.00	0.924	17.72	3	200.33	98.49	151.27	76.60
53........	4.00	0.939	17.71	4	226.75	99.99	157.52	78.61
56........	4.00	0.948	17.70	4	241.25	100.92	160.30	79.31
52........	4.00	0.961	17.66	4	256.25	102.21	162.53	79.58
51........	4.00	0.984	17.52	3	275.00	104.20	166.12	80.42
49........	4.00	0.999	17.53	4	295.00	105.85	169.29	80.63
50........	4.00	1.005	17.51	4	302.00	106.44	169.66	80.45
48........	4.00	1.015	17.48	3	312.67	107.40	169.99	80.02
47........	4.00	1.018	17.51	4	321.75	107.76	165.20	77.38
46........	4.00	1.006	17.54	4	340.75	106.68	144.09	68.05
45........	4.00	0.985	17.60	4	362.50	104.56	109.49	52.58
43........	3.50	0.837	17.84	4	183.75	89.52	140.97	78.01
42........	3.50	0.850	17.83	3	206.33	90.87	149.56	81.58
44........	3.50	0.860	17.76	4	221.50	91.76	153.87	83.44
41........	3.50	0.862	17.78	4	221.75	91.99	154.04	83.23
38........	3.50	0.877	17.79	4	241.50	93.57	160.47	85.20
37........	3.50	0.894	17.72	4	261.50	95.27	165.86	86.83
40........	3.50	0.905	17.67	5	272.80	96.29	168.09	87.31
39........	3.50	0.911	17.67	4	279.75	96.87	168.99	87.25
36........	3.50	0.909	17.69	4	287.75	96.76	165.14	85.26
35........	3.50	0.906	17.69	4	302.25	96.41	155.19	80.42
34........	3.50	0.897	17.71	4	323.00	95.50	136.58	71.37
33........	3.50	0.879	17.75	4	351.50	93.68	106.16	56.42
82........	3.10	0.784	17.88	5	191.20	83.90	138.60	81.65
75........	3.10	0.814	17.75	4	236.75	86.75	154.45	88.65
76........	3.10	0.821	17.73	5	244.80	87.53	156.75	89.26
77........	3.10	0.821	17.73	5	246.80	87.53	157.28	89.57
74........	3.10	0.824	17.74	4	249.75	87.86	158.41	89.82
81........	3.10	0.824	17.78	4	250.00	87.97	158.57	89.59
78........	3.10	0.827	17.73	4	251.00	88.08	158.44	89.67
79........	3.10	0.828	17.72	5	252.40	88.19	158.56	89.67
80........	3.10	0.827	17.75	4	254.25	88.19	158.96	89.74
73........	3.10	0.824	17.74	4	258.75	87.86	156.30	88.62
72........	3.10	0.823	17.75	4	265.75	87.75	152.50	86.53
71........	3.10	0.823	17.76	3	274.00	87.75	148.96	84.47
70........	3.10	0.820	17.77	4	291.50	87.53	140.87	80.04
69........	3.10	0.820	17.77	4	309.50	87.53	130.87	74.36
68........	3.10	0.817	17.80	4	346.25	87.20	104.58	59.54
25........	3.00	0.750	17.92	4	181.00	80.31	130.11	79.90
20........	3.00	0.776	17.87	4	221.25	83.02	147.01	87.58
23........	3.00	0.785	17.83	4	233.25	83.90	150.76	89.07
22........	3.00	0.790	17.83	3	239.00	84.44	153.03	89.83
19........	3.00	0.791	17.85	4	242.00	84.55	153.48	89.88
21........	3.00	0.792	17.83	4	242.50	84.66	153.81	90.05
24........	3.00	0.792	17.81	4	243.00	84.55	152.66	89.59
18........	3.00	0.790	17.85	4	249.50	84.44	150.71	88.37
17........	3.00	0.788	17.86	4	257.25	84.23	147.63	86.73
16........	3.00	0.784	17.86	10	265.10	83.90	144.12	85.00
15........	3.00	0.786	17.84	3	293.67	84.01	133.05	78.45
14........	3.00	0.787	17.84	4	323.25	84.12	117.16	68.99
13........	3.00	0.780	17.87	3	361.00	83.46	87.23	51.69

TABLE 96—Continued.

1	2	3	4	5	6	7	8	9
9........	2.50	0.647	18.12	4	169.75	69.69	114.85	80.37
8........	2.50	0.662	18.07	4	197.00	71.24	124.95	85.78
7........	2.50	0.668	18.06	3	210.67	71.86	127.26	86.66
12........	2.50	0.669	18.03	5	213.20	71.86	128.79	87.85
10........	2.50	0.670	18.04	6	217.50	72.06	128.76	87.53
6........	2.50	0.666	18.07	5	221.80	71.65	127.28	86.88
5........	2.50	0.664	18.07	4	230.00	71.44	125.04	85.60
4........	2.50	0.662	18.07	4	240.00	71.24	123.23	84.60
3........	2.50	0.665	18.05	4	251.75	71.55	121.66	83.25
2........	2.50	0.665	18.07	4	274.50	71.55	116.07	79.34
1........	2.50	0.665	18.05	3	320.00	71.55	96.65	66.14
11........	2.50	0.657	18.10	4	356.50	70.72	64.60	44.60
66........	2.00	0.538	18.26	4	171.50	58.16	98.42	81.90
67........	2.00	0.542	18.24	4	186.25	58.55	102.38	84.72
65........	2.00	0.542	18.25	4	188.75	58.64	102.62	84.74
64........	2.00	0.537	18.28	4	196.75	58.16	101.02	83.98
63........	2.00	0.535	18.29	5	205.40	57.87	99.26	82.88
62........	2.00	0.535	18.29	4	216.50	57.87	98.09	81.90
61........	2.00	0.534	18.30	4	226.00	57.78	95.56	79.87
60........	2.00	0.532	18.31	5	238.40	57.59	93.61	78.45
59........	2.00	0.535	18.31	4	255.75	57.97	92.69	77.18
58........	2.00	0.538	18.32	4	284.75	58.26	86.00	71.21
57........	2.00	0.533	18.33	5	334.40	57.78	60.60	50.57
32........	1.25	0.332	18.53	3	142.00	36.18	55.76	73.50
31........	1.25	0.332	18.65	3	158.33	36.26	57.39	74.99
30........	1.25	0.326	18.66	3	185.00	35.69	55.88	74.15
29........	1.25	0.323	18.65	3	215.00	35.29	51.95	69.76
28........	1.25	0.318	18.65	3	247.33	34.80	44.82	61.03
27........	1.25	0.315	18.65	3	276.33	34.39	33.38	46.00
26........	1.25	0.310	18.65	3	302.33	33.83	18.26	25.58
89........	4.00	0.915	17.65	4	423.00	97.33	....	....
88........	3.50	0.842	17.80	4	426.50	89.86	....	....
87........	3.10	0.751	17.95	3	418.67	80.52	....	....
86........	3.00	0.720	18.02	4	415.75	77.31	....	....
85........	2.50	0.604	18.21	4	401.25	65.23	....	....
84........	2.00	0.482	18.39	4	381.50	52.28	....	....
83........	1.25	0.305	18.70	4	323.50	33.35	....	....

TABLE 97.

Holyoke Tests of a 23-inch Right Hand Wellman-Seaver-Morgan Co. (Runner No. 37) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of opening of the speed-gate. In inches.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
12........	3.30	0.909	17.32	3	184.00	89.68	135.47	76.97
11........	3.30	0.917	17.32	3	199.33	90.44	140.75	79.29
10........	3.30	0.931	17.30	4	217.00	91.74	146.67	81.55
5........	3.30	0.943	17.26	4	229.75	92.82	149.75	82.48
7........	3.30	0.955	17.24	4	245.50	93.97	154.08	83.93
4........	3.30	0.974	17.20	4	263.50	95.75	159.02	85.21
9........	3.30	0.975	17.21	4	264.50	95.82	159.62	85.42
8........	3.30	0.989	17.18	4	276.25	97.14	162.55	85.95
6........	3.30	1.002	17.13	4	288.00	98.24	165.12	86.58
3........	3.30	1.007	17.12	4	299.00	98.71	162.40	84.80
2........	3.30	1.008	17.06	4	314.00	98.63	151.60	79.51
1........	3.30	1.014	17.06	4	334.50	99.25	141.31	73.65
20........	2.89	0.821	17.48	3	178.33	81.37	125.92	78.12
19........	2.89	0.833	17.45	4	193.75	82.41	132.13	81.08
18........	2.89	0.842	17.45	4	208.50	83.30	137.15	83.26
17........	2.89	0.857	17.42	4	226.00	84.79	143.21	85.56
21........	2.89	0.868	17.40	4	237.75	85.84	147.07	86.89
16........	2.89	0.879	17.37	4	251.00	86.81	151.48	88.65
22........	2.89	0.882	17.35	4	261.50	87.04	149.92	87.61
15........	2.89	0.883	17.36	4	270.75	87.19	147.05	85.74
14........	2.89	0.882	17.36	3	288.00	87.04	139.05	81.20
13........	2.89	0.892	17.35	4	314.50	88.09	132.86	76.71
28........	2.47	0.746	17.53	4	190.50	74.03	120.71	82.08
27........	2.47	0.762	17.49	4	214.25	75.54	129.30	86.36
29........	2.47	0.768	17.47	4	223.50	76.05	131.51	87.35
26........	2.47	0.769	17.49	4	230.25	76.19	132.01	87.42
25........	2.47	0.767	17.51	4	238.00	76.05	129.27	85.66
24........	2.47	0.764	17.52	4	247.25	75.76	126.83	84.32
23........	2.47	0.770	17.52	4	260.00	76.34	125.53	82.82
30........	2.47	0.773	17.47	4	278.00	76.55	122.47	80.82
31........	2.47	0.775	17.47	4	294.00	76.77	117.10	77.05
32........	2.47	0.776	17.47	4	312.25	76.84	109.30	71.85
41........	2.06	0.636	17.85	3	181.33	63.64	103.96	80.76
42........	2.06	0.639	17.84	4	191.75	63.98	107.04	82.76
40........	2.06	0.642	17.83	4	200.25	64.25	108.76	83.78
39........	2.06	0.642	17.86	4	211.50	64.25	108.50	83.43
38........	2.06	0.636	17.87	4	219.25	63.70	105.85	82.06
37........	2.06	0.635	17.86	4	230.00	63.57	104.10	80.91
36........	2.06	0.636	17.87	4	243.75	63.70	102.97	79.83
35........	2.06	0.640	17.85	4	261.75	64.04	101.10	78.04
34........	2.06	0.643	17.85	4	281.00	64.39	98.36	75.52
33........	2.06	0.657	17.79	5	310.20	65.62	93.60	70.76
50........	1.65	0.504	18.06	4	192.00	50.79	81.11	78.03
49........	1.65	0.503	18.07	5	204.80	50.66	80.34	77.44
48........	1.65	0.500	18.07	4	216.75	50.34	78.48	76.14
47........	1.65	0.498	18.10	4	230.00	50.21	76.34	74.13
46........	1.65	0.495	18.12	4	244.25	49.96	73.70	71.84
45........	1.65	0.500	18.10	4	267.25	50.40	72.58	70.21
43........	1.65	0.507	18.10	4	288.50	51.10	69.64	66.45
44........	1.65	0.510	18.08	4	305.25	51.36	64.48	61.27

TABLE 97—Continued.

1	2	3	4	5	6	7	8	9
57........	1.24	0.360	18.50	4	181.25	36.71	54.69	71.07
56........	1.24	0.360	18.40	4	197.25	36.60	53.57	70.19
55........	1.24	0.359	18.41	4	221.75	36.48	50.85	66.82
54........	1.24	0.357	18.41	4	244.00	36.25	45.65	60.36
52........	1.24	0.352	18.40	4	258.50	35.80	39.00	52.25
51........	1.24	0.353	18.40	4	274.25	35.85	33.10	44.28
53........	1.24	0.354	18.41	4	292.50	35.97	24.71	32.93
58........	3.30	1.030	17.11	4	470.00	100.91	....	....
59........	2.89	0.957	17.26	5	472.40	94.20	....	....
60........	2.47	0.807	17.55	4	457.25	80.12	....	....

TABLE 98.

Holyoke Tests of a 23-inch Right Hand Wellman-Seaver-Morgan Co. Turbine. Continuation of Test given in Table 97. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate. In inches.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
47........	4.43	1.021	16.48	4	269.00	110.75	146.11	70.59
46........	4.43	1.064	16.38	4	305.50	115.07	147.49	69.00
45........	4.43	1.110	16.30	4	342.50	119.77	144.69	65.35
44........	4.43	1.136	16.23	4	382.50	122.36	115.42	51.25
43........	4.43	1.118	16.28	4	402.25	120.52	72.83	32.73
42........	4.43	1.084	16.40	4	441.25	117.37	....	....
36........	3.95	0.961	16.63	4	269.50	104.79	154.51	78.18
35........	3.95	1.013	16.64	4	312.75	110.43	160.43	76.98
37........	3.95	1.035	16.58	5	332.60	112.62	160.58	75.83
34........	3.95	1.046	16.60	4	347.00	113.92	157.06	73.23
33........	3.95	1.048	16.61	4	362.50	114.17	142.20	66.12
32........	3.95	1.041	16.68	4	383.00	113.60	115.57	53.78
31........	3.95	1.041	16.70	4	404.00	113.68	85.33	39.63
30........	3.95	1.027	16.77	4	426.75	112.46	51.51	24.08
41........	3.82	0.958	16.72	4	297.00	104.71	161.31	81.25
40........	3.82	0.980	16.69	4	316.00	107.03	162.10	80.01
38........	3.82	0.990	16.66	5	329.20	108.00	158.94	77.89
39........	3.82	0.994	16.65	4	345.50	108.40	145.95	71.31
29........	3.58	0.957	16.42	4	421.25	103.67	50.84	26.34
28........	3.58	0.976	16.41	4	399.25	105.66	84.33	42.89
27........	3.58	0.964	16.41	4	373.00	104.39	112.55	57.93
26........	3.58	0.940	16.47	4	330.00	102.01	139.40	73.16
25........	3.58	0.943	16.46	4	315.25	102.25	152.20	79.74
24........	3.58	0.926	16.51	4	291.00	100.51	158.05	83.99
23........	3.58	0.878	16.63	4	243.75	95.67	147.10	81.53
22........	3.58	0.846	16.71	4	207.50	92.43	137.75	78.64
9........	3.30	0.833	16.85	4	223.75	91.36	141.78	81.21
8........	3.30	0.849	16.80	4	244.00	93.05	147.26	83.06
10........	3.30	0.872	16.74	4	267.75	95.36	153.51	84.79
2........	3.30	0.894	16.79	5	291.20	97.92	158.16	84.83
7........	3.30	0.899	16.65	4	308.50	98.00	148.94	80.49
1........	3.30	0.900	16.85	4	328.00	98.71	138.56	73.46
4........	3.30	0.912	16.75	4	349.75	99.73	126.64	66.85
3........	3.30	0.923	16.75	5	371.60	100.98	112.13	58.45
5........	3.30	0.930	16.70	4	389.00	101.54	93.90	48.83
6........	3.30	0.933	16.67	4	406.00	101.85	73.51	38.17
11........	3.30	0.932	16.65	4	423.50	101.69	51.12	26.62
21........	2.89	0.736	16.96	4	185.00	81.00	122.81	78.83
20........	2.89	0.766	16.91	4	227.50	84.20	137.30	85.03
19........	2.89	0.786	16.86	4	261.25	86.29	141.90	86.00
18........	2.89	0.783	16.91	4	278.75	86.06	134.58	81.54
17........	2.89	0.793	16.88	5	302.60	87.04	127.83	76.72
16........	2.89	0.803	16.84	4	329.00	88.09	119.13	70.81
15........	2.89	0.810	16.83	4	351.50	88.85	106.06	62.54
14........	2.89	0.811	16.84	4	371.75	89.00	89.74	52.80
13........	2.89	0.807	16.84	4	390.75	88.55	70.74	41.83
12........	2.89	0.824	16.83	4	417.75	90.36	50.42	29.24
48........	3.95	1.013	16.49	4	449.50	109.94	....	....
49........	3.58	0.937	16.69	4	453.00	102.33	....	....
50........	3.30	0.908	16.74	4	460.00	99.25	....	....

TABLE 99.

Holyoke Tests of a 30-inch Right Hand Allis-Chalmers Co. (Type No. 13) Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of opening of the speed-gate. In inches.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
35........	4.10	0.972	15.90	4	171.00	145.81	207.60	78.96
34........	4.10	0.984	15.86	4	188.00	147.40	211.94	79.94
33........	4.10	0.990	15.82	4	197.00	148.06	212.97	80.17
32........	4.10	0.997	15.78	4	206.50	149.00	214.88	80.59
31........	4.10	1.004	15.75	4	212.75	149.80	215.24	80.44
30........	4.10	1.011	15.71	4	219.50	150.74	215.72	80.32
36........	4.10	1.013	15.84	5	225.40	151.55	215.01	78.98
29........	4.10	1.025	15.67	4	232.25	152.62	214.83	79.21
37........	4.10	1.025	15.82	4	237.75	153.30	213.04	77.46
38........	4.10	1.031	15.81	4	242.75	154.11	210.51	76.18
28........	4.10	1.036	15.67	4	264.00	154.24	183.15	66.82
39........	4.10	0.989	15.99	4	325.25	148.73	....	....
19........	3.75	0.907	14.70	4	137.25	130.71	169.01	77.56
20........	3.75	0.906	14.45	4	137.25	129.57	165.04	77.73
21........	3.75	0.910	14.29	4	144.00	129.31	166.50	79.45
18........	3.75	0.924	15.36	4	170.50	136.11	197.14	83.15
17........	3.75	0.933	15.36	4	183.25	137.54	201.29	84.01
16........	3.75	0.944	15.37	4	198.00	139.24	206.04	84.89
22........	3.75	0.953	15.08	4	206.25	139.24	202.70	85.12
15........	3.75	0.960	15.35	4	214.00	141.46	210.32	85.41
23........	3.75	0.958	14.92	4	209.75	139.11	200.08	85.00
24........	3.75	0.962	14.70	4	211.00	138.71	195.17	84.40
25........	3.75	0.970	14.48	5	219.40	138.85	190.26	83.44
26........	3.75	0.970	14.32	4	225.50	138.06	182.51	81.40
27........	3.75	0.959	14.27	4	234.50	136.24	162.68	73.78
14........	3.75	0.942	15.56	4	326.50	139.76	....	....
11........	3.375	0.846	16.40	4	145.25	128.80	188.94	78.87
10........	3.375	0.857	16.41	4	163.50	130.59	203.22	83.62
9........	3.375	0.867	16.40	4	176.75	131.99	209.47	85.33
8........	3.375	0.872	16.39	4	184.75	132.76	213.61	86.56
7........	3.375	0.878	16.39	4	193.50	133.66	218.14	87.80
5........	3.375	0.881	16.40	3	200.00	134.18	219.68	88.03
12........	3.375	0.887	16.24	4	204.25	134.44	220.81	89.18
6........	3.375	0.886	16.38	4	206.75	134.82	221.12	88.29
4........	3.375	0.891	16.33	4	214.75	135.47	223.47	89.07
13........	3.375	0.893	16.19	4	212.00	135.08	220.61	88.95
3........	3.375	0.889	16.34	4	219.50	135.21	215.72	86.09
2........	3.375	0.887	16.22	5	225.00	134.31	208.12	84.24
1........	3.375	0.877	16.26	5	237.00	133.02	191.82	78.20
93........	3.21	0.825	16.50	3	160.00	126.03	196.10	83.15
92........	3.21	0.837	16.50	4	174.75	127.79	207.10	86.61
91........	3.21	0.847	16.50	4	189.50	129.31	213.63	88.29
90........	3.21	0.851	16.47	4	196.75	129.95	216.11	89.04
89........	3.21	0.856	16.42	4	203.00	130.46	217.11	89.37
88........	3.21	0.855	16.42	4	211.25	130.33	213.72	88.05
87........	3.21	0.852	16.42	4	216.50	129.82	206.52	85.43
86........	3.21	0.849	16.40	4	222.75	129.31	199.60	82.99
84........	3.21	0.843	16.35	3	233.00	128.17	188.58	79.35
85........	3.21	0.839	16.37	4	240.25	127.67	180.56	76.18

TABLE 99—Continued.

1	2	3	4	5	6	7	8	9
52........	3.125	0.802	16.59	3	149.67	122.90	186.03	80.45
51........	3.125	0.816	16.58	4	169.00	125.02	200.29	85.20
50........	3.125	0.826	16.56	4	181.50	126.41	207.76	87.51
49........	3.125	0.834	16.64	4	195.75	127.92	215.02	89.07
48........	3.125	0.838	16.61	3	202.00	128.42	216.04	89.31
47........	3.125	0.837	16.65	4	206.75	128.42	215.14	88.72
46........	3.125	0.836	16.66	5	210.00	128.30	212.46	87.64
45........	3.125	0.829	16.69	4	217.50	127.41	204.96	84.99
44........	3.125	0.826	16.71	4	223.75	127.04	200.50	83.28
43........	3.125	0.822	16.74	4	235.00	126.53	190.20	79.18
42........	3.125	0.822	16.75	4	248.50	126.53	179.58	74.71
41........	3.125	0.830	16.78	4	263.25	127.79	167.41	68.84
40........	3.125	0.812	16.86	5	334.60	125.40	....	....
62........	2.82	0.742	16.68	2	163.00	114.03	180.93	83.88
60........	2.82	0.747	16.71	3	173.00	114.77	188.03	86.45
59........	2.82	0.756	16.79	4	185.00	116.48	195.72	88.24
61........	2.82	0.756	16.66	4	187.00	116.11	193.51	88.21
58........	2.82	0.753	16.90	4	192.75	116.48	195.01	87.35
57........	2.82	0.750	16.84	4	196.75	115.74	191.09	86.45
56........	2.82	0.747	16.85	4	202.50	115.25	187.31	85.05
55........	2.82	0.743	16.86	4	211.00	114.77	182.96	83.38
54........	2.82	0.740	16.88	4	220.00	114.40	178.06	81.31
53........	2.82	0.751	16.90	4	244.00	116.11	169.27	76.06
72........	2.40	0.636	17.04	3	148.00	98.72	149.73	78.49
73........	2.40	0.647	16.97	4	159.25	100.22	157.43	81.62
71........	2.40	0.654	17.01	3	171.00	101.51	165.09	84.31
70........	2.40	0.653	17.04	4	178.25	101.39	164.88	84.15
69........	2.40	0.650	17.06	4	182.00	100.92	163.09	83.53
68........	2.40	0.648	17.06	4	186.50	100.69	161.73	83.02
67........	2.40	0.644	17.07	4	195.50	100.11	158.23	81.65
66........	2.40	0.642	17.11	4	204.75	99.87	153.88	79.41
65........	2.40	0.635	17.12	4	218.75	98.83	145.43	75.79
63........	2.40	0.633	17.10	4	233.25	98.37	134.85	70.69
64........	2.40	0.633	17.10	4	247.50	98.37	121.62	63.75
83........	2.00	0.549	17.20	3	152.33	85.65	129.45	77.48
82........	2.00	0.553	17.21	4	160.75	86.20	131.96	78.44
81........	2.00	0.552	17.18	4	168.75	85.98	131.70	78.62
80........	2.00	0.552	17.14	3	181.00	85.98	130.80	78.26
79........	2.00	0.548	17.16	3	191.00	85.43	126.98	76.38
78........	2.00	0.539	17.17	4	205.00	84.01	118.51	72.45
77........	2.00	0.534	17.18	4	219.50	83.24	107.86	66.51
75........	2.00	0.531	17.24	3	232.00	82.91	93.89	57.92
76........	2.00	0.530	17.23	4	240.25	82.69	83.34	51.58
74........	2.00	0.513	17.36	3	282.00	80.42	....	....

TABLE 100.

Holyoke Tests of a 30-inch Left Hand Smith (Type R) Turbine. Swing-gate. Long conical Draft-tube.

Number of the experiment.	Proportional part of		Head acting on the wheel.	Duration of the experiment.	Revolutions of the wheel.	Quantity of water discharged by the wheel.	Power developed by the wheel.	Efficiency of the wheel.
	the full opening of the speed-gate.	the full discharge of the wheel. Per cent.	In feet.	In min.	Per min.	Cubic ft. per sec.	H. P.	In per cent.
1	2	3	4	5	6	7	8	9
98........	1.000	0.798	16.62	6	Still	127.67	....	....
20........	1.000	0.958	16.01	3	143.33	150.47	223.10	81.66
19........	1.000	0.966	15.97	4	156.25	151.41	229.70	83.76
18........	1.000	0.975	15.92	3	170.00	152.62	235.21	85.36
17........	1.000	0.990	15.88	4	186.25	154.79	241.59	86.66
16........	1.000	1.006	15.81	5	201.60	156.95	244.06	86.73
15........	1.000	1.023	15.69	4	215.25	159.00	241.98	85.53
14........	1.000	1.031	15.63	4	226.50	159.95	235.04	82.90
13........	1.000	1.030	15.60	4	234.25	159.68	222.82	78.87
12........	1.000	1.016	15.63	4	241.00	157.64	201.46	72.10
11........	1.000	1.002	15.60	4	246.00	155.33	177.27	64.51
10........	1.000	1.070	15.38	7	327.71	164.62	....	....
30........	0.955	0.910	16.21	4	141.00	143.69	215.41	81.55
29........	0.955	0.919	16.16	4	156.50	145.01	225.55	84.87
28........	0.955	0.931	16.12	4	171.75	146.61	232.68	86.81
27........	0.955	0.940	16.09	4	182.75	147.93	237.05	87.82
26........	0.955	0.950	16.05	4	193.50	149.27	239.84	88.27
31........	0.955	0.956	16.07	4	202.25	150.34	241.35	88.09
25........	0.955	0.964	16.01	3	209.00	151.28	240.97	87.73
24........	0.955	0.969	16.01	4	219.75	152.08	234.37	84.88
23........	0.955	0.962	16.03	4	225.50	151.14	221.00	80.43
22........	0.955	0.954	16.03	4	251.25	149.94	173.81	63.76
21........	0.955	1.020	15.83	4	332.00	159.27	....	....
60........	0.879	0.859	16.42	4	141.25	136.50	209.28	82.33
59........	0.879	0.870	16.35	4	156.25	138.06	220.69	86.21
57........	0.879	0.883	16.32	4	174.25	139.89	231.04	89.24
56........	0.879	0.891	16.27	4	184.25	141.07	233.68	89.77
61........	0.879	0.896	16.26	4	188.75	141.72	235.04	89.94
58........	0.879	0.896	16.27	4	190.25	141.85	235.81	90.09
54........	0.879	0.901	16.24	4	194.75	142.51	235.77	89.83
53........	0.879	0.903	16.23	4	202.25	142.77	233.19	88.74
55........	0.879	0.903	16.29	3	208.00	143.03	231.43	87.58
52........	0.879	0.898	16.23	3	211.00	141.99	225.03	86.10
51........	0.879	0.891	16.23	4	215.50	140.80	211.20	81.49
50........	0.879	0.889	16.19	4	223.50	140.28	199.71	77.54
49........	0.879	0.891	16.17	4	237.25	140.54	184.64	71.64
48........	0.879	0.925	16.10	4	265.40	145.68	168.30	63.27
47........	0.879	0.948	16.08	4	333.00	149.13	....	....
81........	0.802	0.809	16.71	4	144.00	129.82	203.39	82.67
80........	0.802	0.818	16.65	4	156.75	130.97	212.36	85.87
78........	0.802	0.828	16.52	4	168.25	131.99	218.24	88.25
77........	0.802	0.836	16.43	4	178.75	132.89	221.55	89.47
79........	0.802	0.837	16.49	4	184.50	133.40	223.36	89.54
76........	0.802	0.839	16.41	4	188.00	133.40	222.18	89.49
75........	0.802	0.839	16.44	4	195.25	133.40	219.50	88.25
74........	0.802	0.832	16.46	4	199.00	132.50	212.24	85.81
73........	0.802	0.828	16.45	4	203.50	131.74	205.30	83.53
72........	0.802	0.828	16.44	4	212.75	131.74	196.25	79.89
71........	0.802	0.834	16.44	4	229.75	132.63	185.43	74.99
82........	0.802	0.856	16.48	5	250.00	136.37	180.16	70.68
70........	0.802	0.856	16.40	5	253.40	135.98	175.30	69.31

TABLE 100—Continued.

1	2	3	4	5	6	7	8	9
92........	0.726	0.749	16.80	4	145.75	120.42	189.05	82.40
91........	0.726	0.758	16.76	4	160.25	121.78	198.62	85.81
90........	0.726	0.765	16.72	4	172.25	122.78	203.57	87.44
89........	0.726	0.768	16.77	4	182.00	123.40	204.60	87.18
88........	0.726	0.765	16.75	4	187.25	122.78	199.71	85.62
87........	0.726	0.759	16.74	4	191.25	121.78	192.94	83.46
86........	0.726	0.756	16.74	4	201.50	121.40	185.86	80.64
85........	0.726	0.764	16.72	4	216.50	122.65	180.98	77.82
84........	0.726	0.772	16.71	4	237.50	123.90	171.15	72.89
83........	0.726	0.787	16.67	4	257.75	126.15	156.01	65.42
68........	0.649	0.662	16.93	3	145.67	106.80	163.76	79.86
69........	0.649	0.670	17.01	3	159.33	108.36	172.68	82.61
67........	0.649	0.674	16.97	4	169.00	108.95	175.37	83.64
66........	0.649	0.668	16.96	3	178.00	108.00	169.32	81.51
65........	0.649	0.665	16.97	3	189.00	107.52	163.44	78.98
64........	0.649	0.661	16.97	4	209.50	106.92	150.97	73.37
62........	0.649	0.662	17.00	4	230.50	107.16	132.88	64.32
63........	0.649	0.695	16.90	4	267.25	112.09	107.85	50.20
45........	0.578	0.596	17.05	4	147.00	96.53	144.07	77.18
44........	0.578	0.601	17.02	4	160.75	97.22	148.27	79.01
46........	0.578	0.599	17.05	4	166.25	96.99	146.64	78.19
43........	0.578	0.598	17.03	5	174.80	96.87	146.12	78.10
42........	0.578	0.595	17.07	3	187.00	96.41	140.15	75.09
41........	0.578	0.592	17.09	4	205.25	96.07	130.16	69.90
40........	0.578	0.593	17.04	4	224.00	96.07	116.22	62.60
39........	0.578	0.596	17.02	5	249.00	96.53	86.13	46.22
38........	0.478	0.525	17.18	4	144.50	85.43	124.96	75.07
37........	0.478	0.524	17.20	4	154.25	85.32	124.49	74.80
36........	0.478	0.525	17.13	4	163.75	85.32	122.72	74.04
35........	0.478	0.520	17.16	4	177.00	84.55	117.35	71.32
34........	0.478	0.519	17.17	4	192.50	84.34	110.98	67.57
32........	0.478	0.523	17.23	4	221.75	85.21	95.88	57.58
33........	0.478	0.525	17.18	4	249.50	85.43	57.53	34.57
7........	0.382	0.429	17.36	4	142.00	70.20	98.23	71.08
6........	0.382	0.426	17.36	4	165.00	69.69	95.12	69.33
8........	0.382	0.418	17.35	4	186.75	68.37	80.75	60.02
5........	0.382	0.424	17.35	4	215.25	69.28	62.04	45.51
9........	0.382	0.427	17.37	4	239.75	69.79	34.55	25.13
4........	0.382	0.430	17.28	4	251.00	70.10	....	....
97........	0.304	0.316	17.86	4	152.75	52.47	66.04	62.14
96........	0.304	0.315	17.87	4	178.00	52.19	56.44	53.36
95........	0.304	0.315	17.87	4	199.75	52.19	40.30	38.11
94........	0.304	0.318	17.92	4	221.75	52.75	19.18	17.89
93........	0.304	0.317	17.91	4	225.25	52.66	....	....
3........	0.191	0.236	17.80	5	196.00	39.00	....	....
2........	0.143	0.170	17.91	4	169.25	28.15	....	....
1........	0.095	0.111	18.15	4	134.75	18.50	....	....

TABLE 101.

Holyoke Tests of a 30-inch Right Hand Improved Special Samson Turbine. Swing-gate. Conical Draft-tube.

Number of the experiment.	Proportional part of the full opening of the speed-gate.	Proportional part of the full discharge of the wheel. Per cent.	Head acting on the wheel. In feet.	Duration of the experiment. In min.	Revolutions of the wheel. Per min.	Quantity of water discharged by the wheel. Cubic ft. per sec.	Power developed by the wheel. H. P.	Efficiency of the wheel. In per cent.
1	2	3	4	5	6	7	8	9
26........	1.000	0.965	16.16	4	172.75	180.49	269.65	81.58
25........	1.000	0.983	16.08	4	185.25	183.36	278.45	83.34
24........	1.000	0.994	16.04	4	195.25	185.23	282.19	83.82
23........	1.000	1.003	16.03	3	204.67	186.82	283.97	83.68
22........	1.000	1.010	15.98	4	209.50	187.69	284.62	83.74
21........	1.000	1.016	15.97	4	214.00	188.85	284.55	83.26
20........	1.000	1.023	15.97	3	219.00	190.16	284.87	82.78
19........	1.000	1.034	15.93	4	228.75	191.92	284.32	82.07
18........	1.000	1.051	15.84	4	242.50	194.55	280.39	80.29
17........	1.000	1.061	15.84	5	256.20	196.32	266.60	75.66
38........	0.891	0.894	16.40	3	163.33	168.36	261.55	83.59
37........	0.891	0.917	16.27	4	180.25	172.00	276.14	87.08
36........	0.891	0.928	16.24	4	189.75	173.97	279.73	87.37
35........	0.891	0.940	16.21	3	200.00	176.08	283.28	87.58
34........	0.891	0.950	16.21	3	210.00	177.92	285.30	87.29
32........	0.891	0.956	16.15	4	214.25	178.64	284.88	87.14
31........	0.891	0.962	16.14	4	220.25	179.78	286.49	87.13
33........	0.891	0.966	16.13	4	223.75	180.35	284.58	86.33
30........	0.891	0.971	16.09	6	229.33	181.06	285.04	86.34
29........	0.891	0.975	16.08	4	238.00	181.78	275.18	83.08
28........	0.891	0.984	16.06	4	251.00	183.36	261.19	78.27
27........	0.891	0.979	16.15	5	264.00	182.93	228.93	68.38
49........	0.796	0.838	16.52	3	164.67	158.32	257.04	86.72
48........	0.796	0.850	16.49	3	177.00	160.50	266.05	88.71
47........	0.796	0.862	16.45	4	187.50	162.56	270.99	89.43
46........	0.796	0.873	16.41	4	199.75	164.48	277.15	90.61
45........	0.796	0.877	16.39	4	204.25	165.04	277.49	90.53
44........	0.796	0.877	16.41	4	207.75	165.31	276.24	89.86
43........	0.796	0.878	16.43	4	213.75	165.45	271.86	88.25
42........	0.796	0.882	16.47	3	221.00	166.42	268.30	86.38
41........	0.796	0.886	16.46	3	231.33	167.25	260.78	83.59
40........	0.796	0.881	16.48	4	237.00	166.28	246.62	79.42
39........	0.796	0.887	16.47	4	249.25	167.39	230.55	73.80
71........	0.749	0.789	16.74	4	153.00	150.21	243.24	85.36
70........	0.749	0.806	16.70	3	171.00	153.16	257.03	88.68
69........	0.749	0.819	16.67	3	183.00	155.46	264.49	90.06
68........	0.749	0.822	16.64	5	188.80	156.00	267.41	90.91
67........	0.749	0.825	16.63	4	191.75	156.41	266.05	90.26
66........	0.749	0.827	16.60	3	196.00	156.68	266.28	90.35
65........	0.749	0.828	16.59	4	199.00	156.82	264.60	89.75
64........	0.749	0.827	16.61	4	202.25	156.68	263.08	89.21
63........	0.749	0.829	16.62	4	206.25	157.09	262.32	88.66
62........	0.749	0.834	16.63	4	214.25	158.18	260.11	87.26
61........	0.749	0.835	16.66	5	220.20	158.45	254.60	85.11
60........	0.749	0.829	16.65	4	225.75	157.23	241.44	81.39
59........	0.749	0.834	16.65	5	236.60	158.18	232.53	77.91
80........	0.700	0.747	16.90	4	148.00	142.90	231.01	84.41
79........	0.700	0.761	16.86	4	166.00	145.41	234.72	88.09
78........	0.700	0.772	16.85	4	177.50	147.40	251.41	89.33
77........	0.700	0.775	16.83	4	183.75	147.93	254.95	90.37
76........	0.700	0.778	16.85	4	187.25	148.46	254.40	89.74
75........	0.700	0.778	16.83	4	194.00	148.46	252.35	89.12
74........	0.700	0.783	16.81	4	201.50	149.27	250.45	88.08
73........	0.700	0.782	16.80	3	210.67	149.13	243.58	85.80
72........	0.700	0.789	16.81	4	231.75	150.47	227.76	79.46

TABLE 101—Continued.

1	2	3	4	5	6	7	8	9
89........	0.600	0.658	17.18	3	153.33	126.91	208.31	84.31
88........	0.600	0.665	17.15	3	164.00	128.17	215.22	86.40
86........	0.600	0.670	17.11	3	170.67	128.80	217.07	86.92
87........	0.600	0.671	17.12	4	175.75	129.06	218.45	87.25
85........	0.600	0.672	17.13	4	180.50	129.44	219.13	87.21
84........	0.600	0.670	17.17	3	186.00	129.18	215.06	85.56
83........	0.600	0.669	17.17	4	192.50	128.93	211.45	84.29
82........	0.600	0.670	17.18	3	202.00	129.06	204.36	81.34
81........	0.600	0.675	17.18	5	223.00	130.08	193.38	76.36
96........	0.500	0.560	17.48	3	149.00	108.83	172.28	79.92
95........	0.500	0.564	17.51	4	157.25	109.67	177.29	81.46
97........	0.500	0.565	17.46	3	161.67	109.79	177.58	81.75
94........	0.500	0.564	17.51	3	166.00	109.79	177.54	81.50
93........	0.500	0.565	17.51	3	174.67	109.91	176.71	81.03
92........	0.500	0.564	17.50	3	187.00	109.79	172.97	79.45
91........	0.500	0.565	17.50	3	203.00	109.91	164.30	75.38
90........	0.500	0.572	17.50	3	223.33	111.24	154.93	70.23

APPENDIX C

COEFFICIENTS FOR WEIRS OF VARIOUS SHAPES.

These figures are reduced directly from the diagrams of Mr. George W. Rafter in the Report of the Board of Engineers of Deep Waterways, 1900.

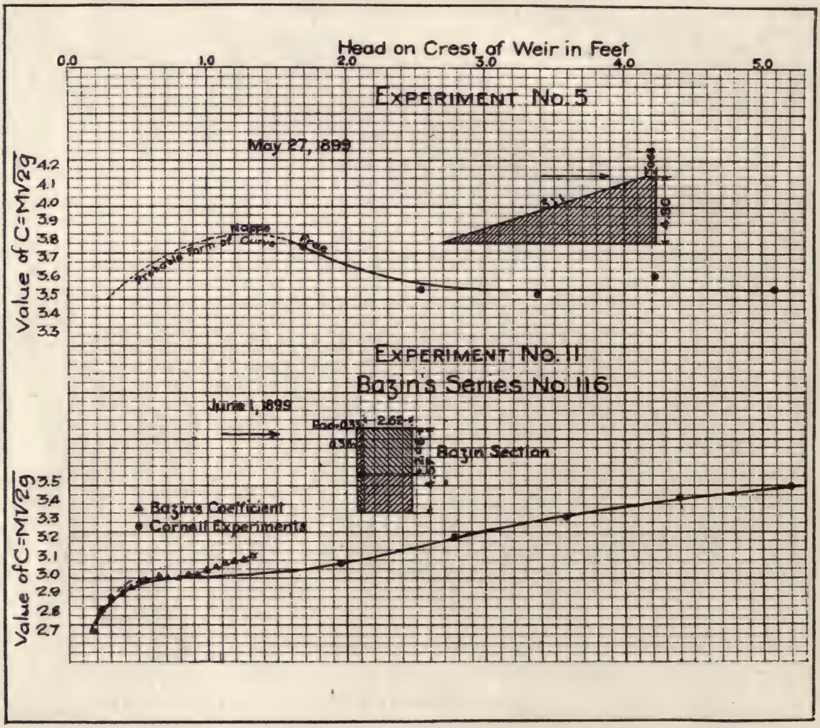


Fig. 431.—Weir Coefficients for Weirs of Various Shapes.

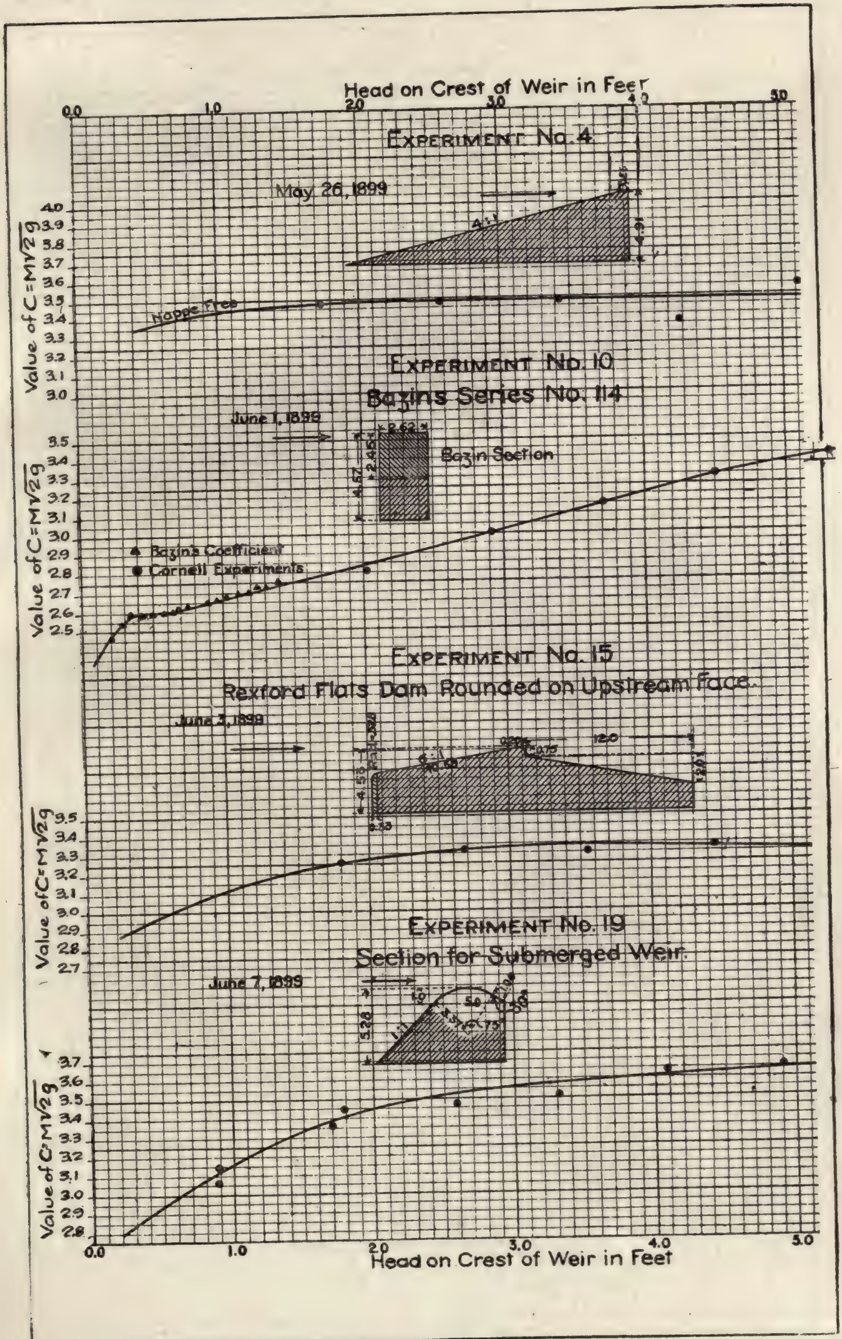


Fig. 432.—Weir Coefficients for Weirs of Various Shapes.

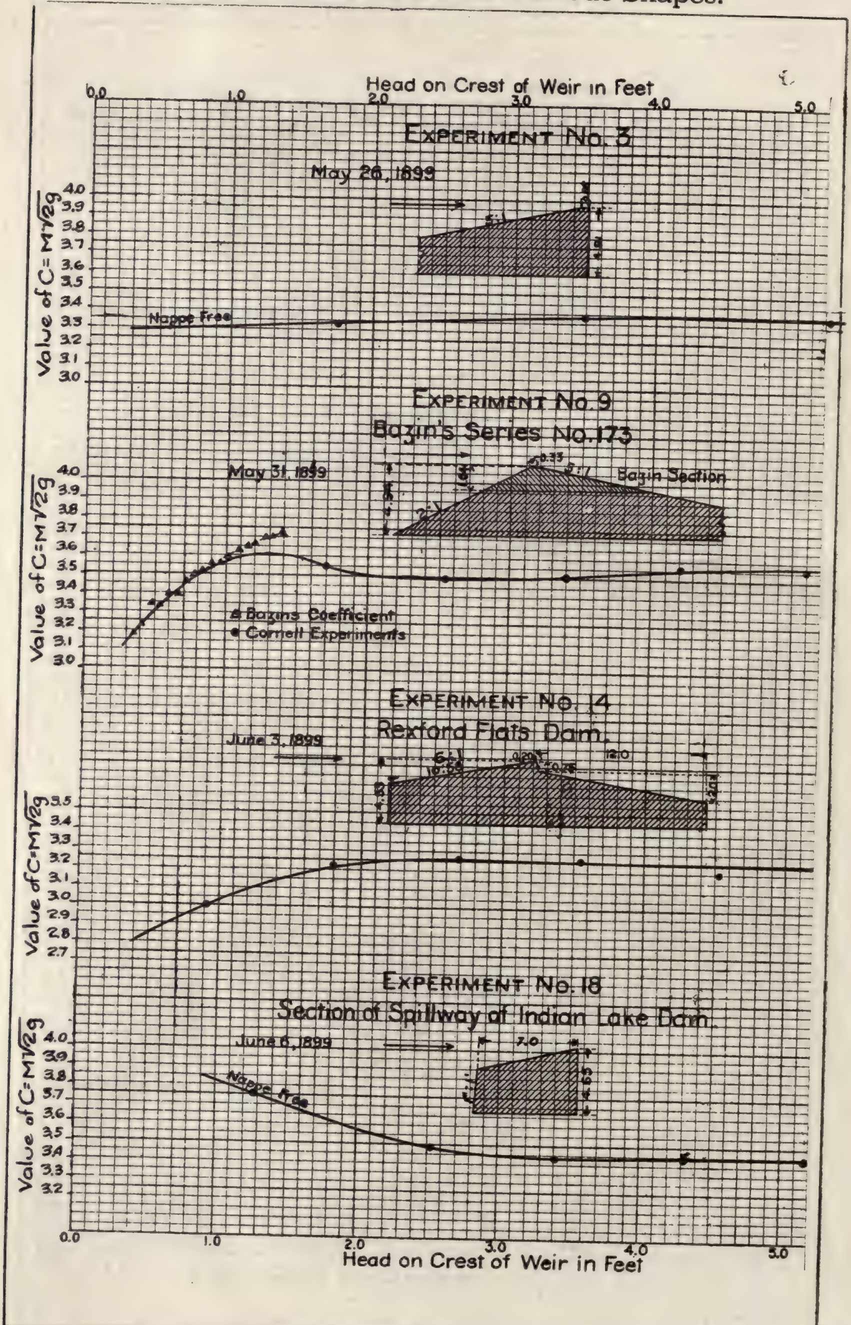


Fig. 433.—Weir Coefficients for Weirs of Various Shapes.

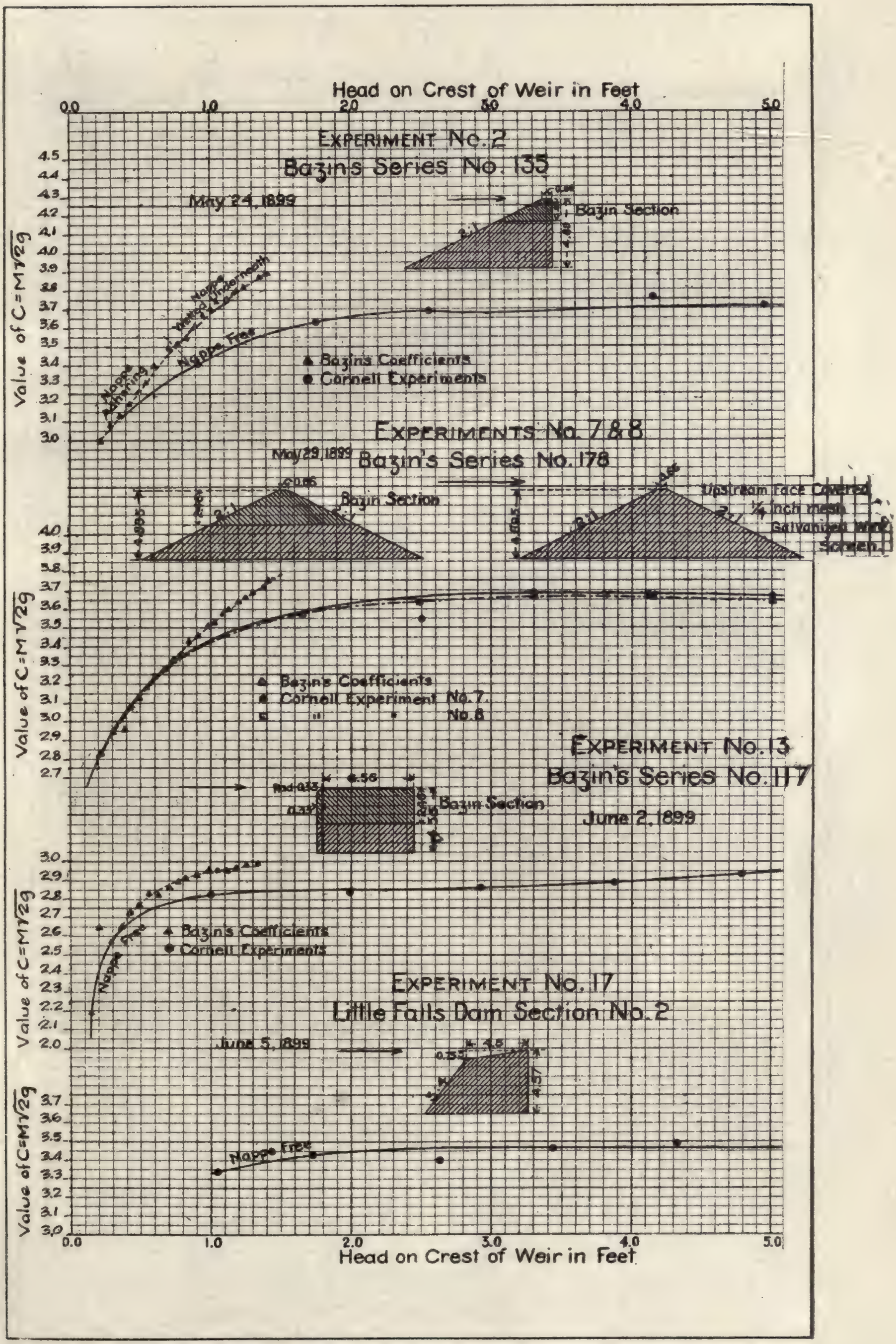


Fig. 434.—Weir Coefficients for Weirs of Various Shapes.

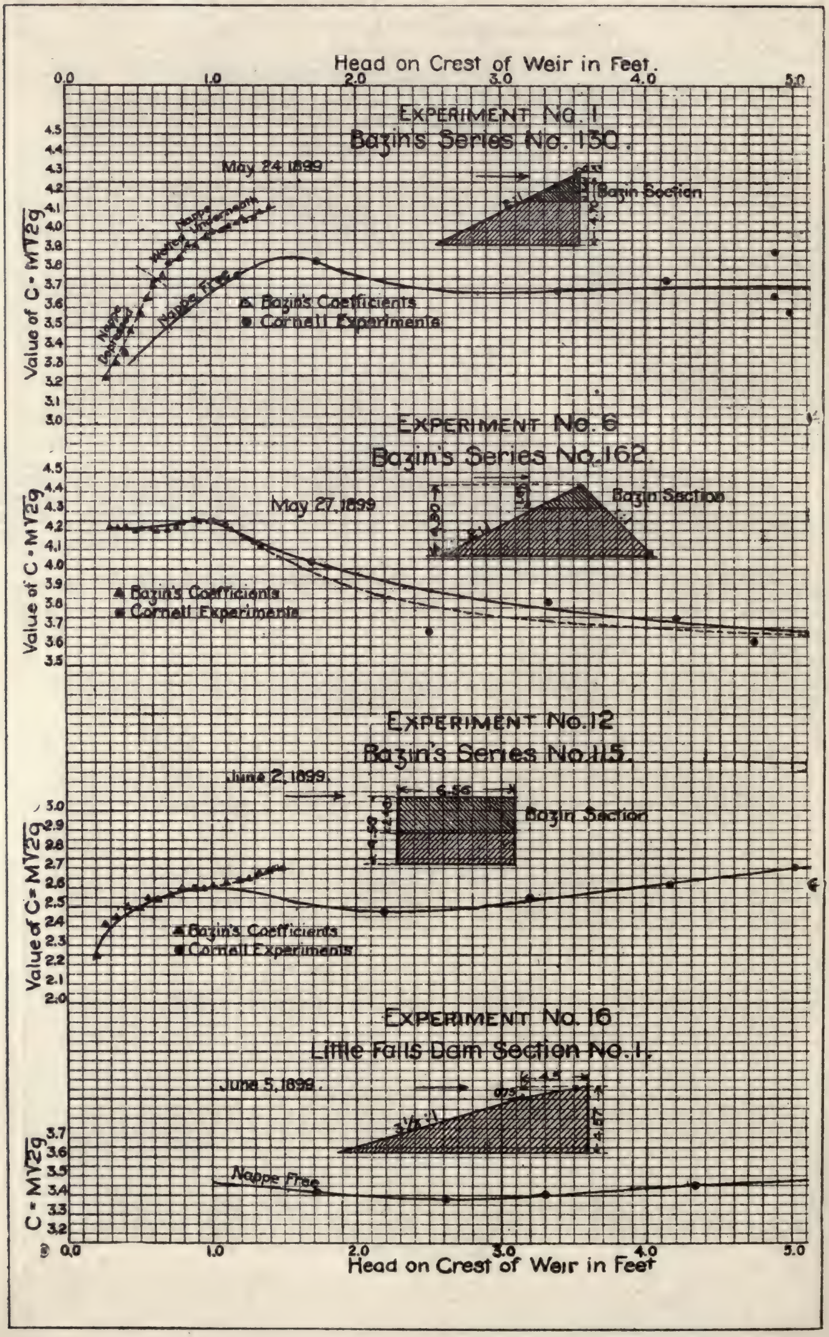
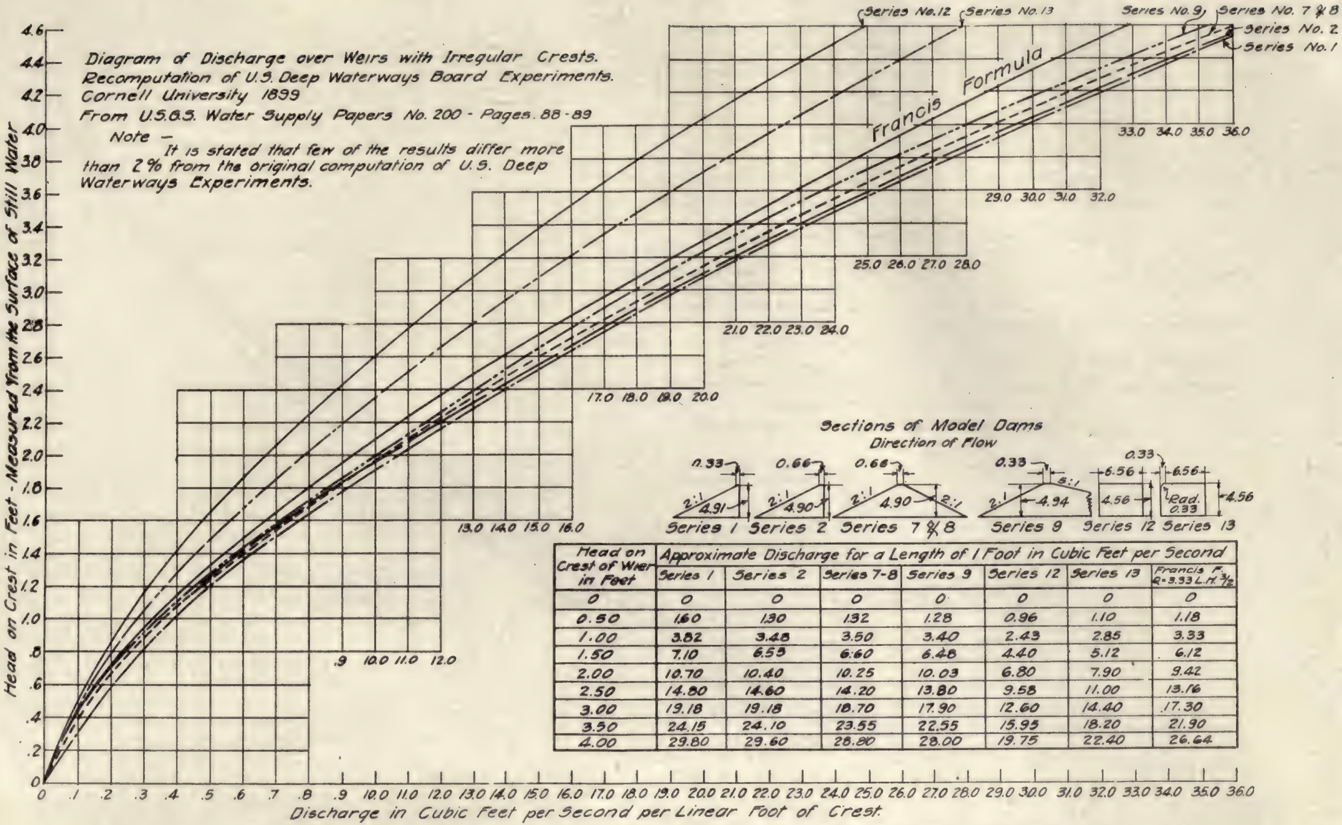


Fig. 435.—Weir Coefficients for Weirs of Various Shapes.



Head on Crest of Wier in Feet	Approximate Discharge for a Length of 1 Foot in Cubic Feet per Second						
	Series 1	Series 2	Series 7-8	Series 9	Series 12	Series 13	Francis F. $Q=3.33\,L.H.^{3/2}$
0	0	0	0	0	0	0	0
0.50	1.60	1.30	1.32	1.28	0.96	1.10	1.18
1.00	3.82	3.48	3.50	3.40	2.43	2.85	3.33
1.50	7.10	6.55	6.60	6.48	4.40	5.12	6.12
2.00	10.70	10.40	10.25	10.03	6.80	7.90	9.42
2.50	14.80	14.60	14.20	13.80	9.58	11.00	13.16
3.00	19.18	19.18	18.70	17.90	12.60	14.40	17.30
3.50	24.15	24.10	23.55	22.55	15.95	18.20	21.90
4.00	29.80	29.60	28.80	28.00	19.75	22.40	26.64

Fig. 436.—Comparison of Bazin's Experimental Results with Francis's Formula.

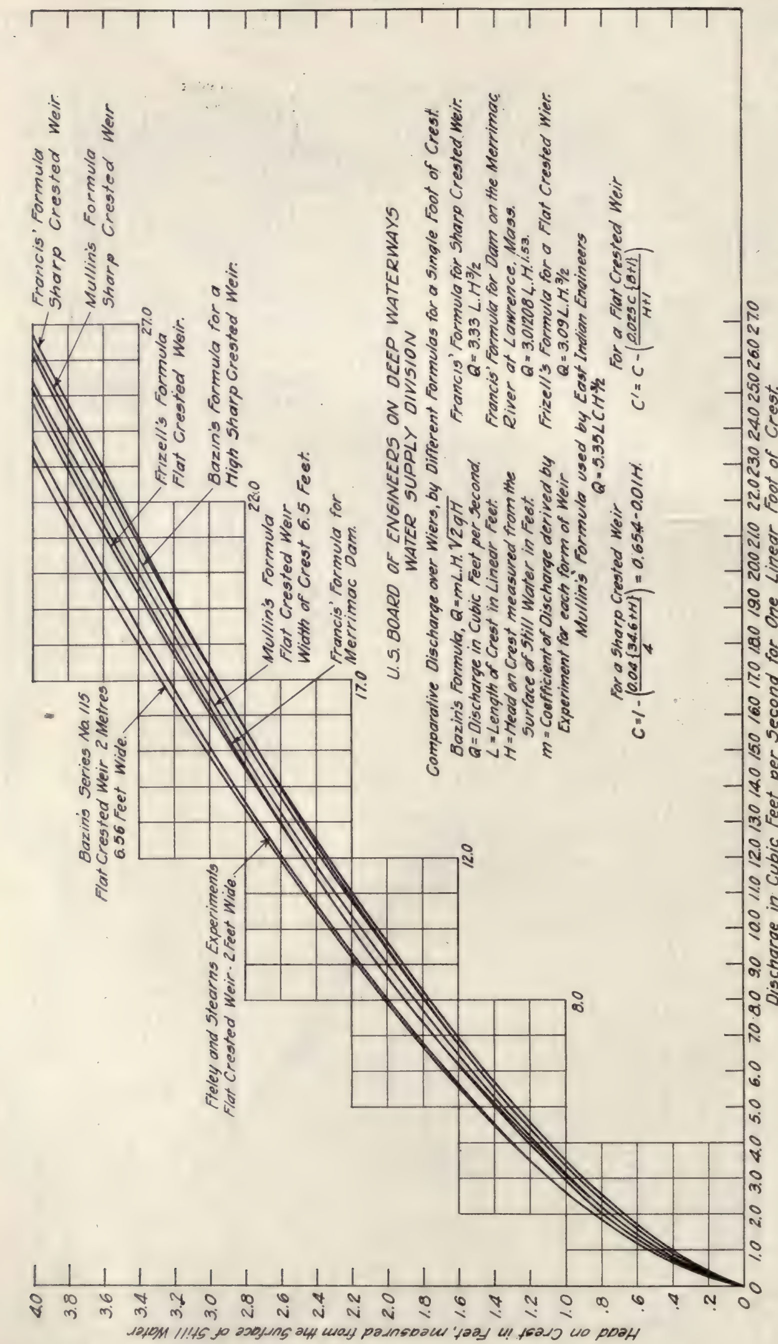


Fig. 437.—Comparison of Various Weir Formulas.

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